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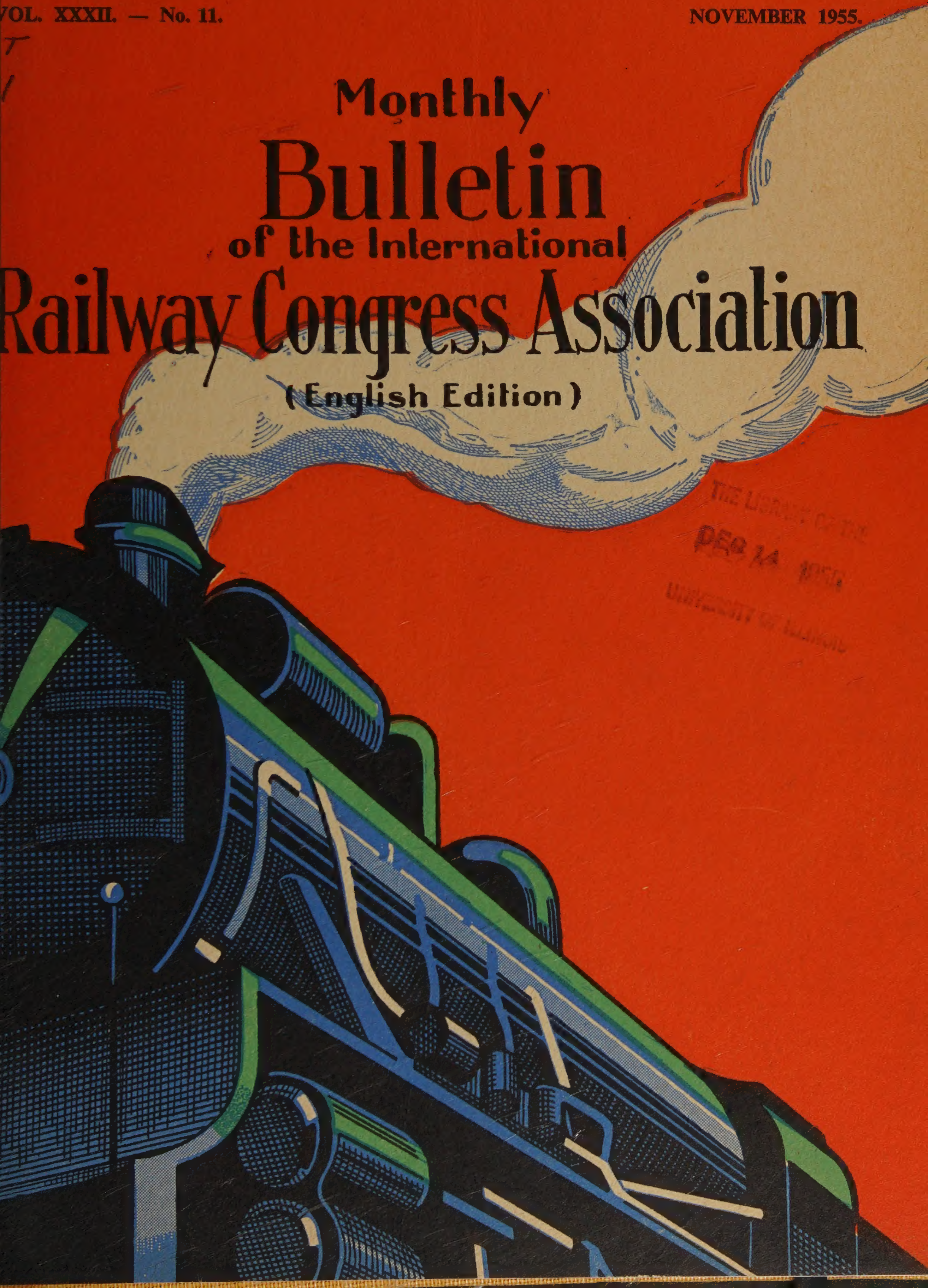
(English Edition)



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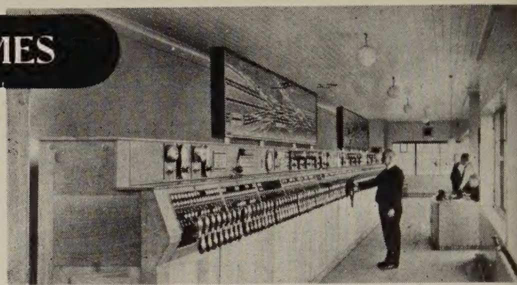

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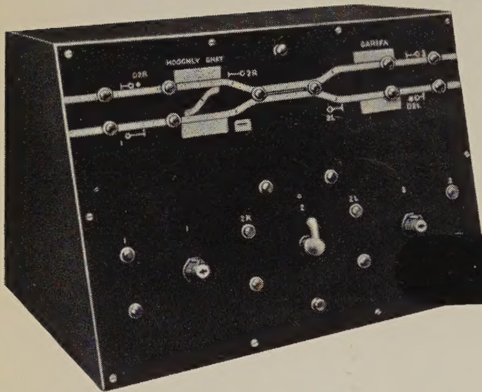

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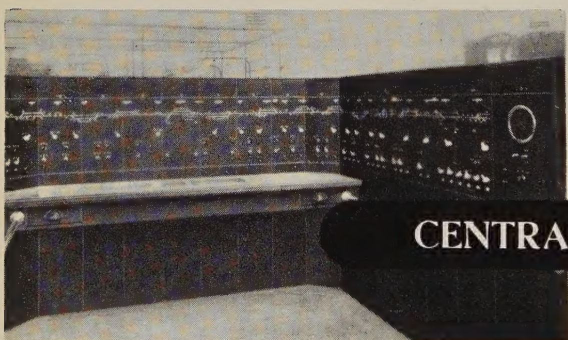

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Bulletin of the International Railway Congress Association

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BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[621 .431 .72 (81)]

Diesel-hydraulic C'C' locomotive of 1 900 HP for metre gauge line,

by Richard SCHEFFER, Dipl.-Eng., and Walter MÜLLER, Dipl.-Eng., Esslingen (Chapters I to III, VI and VII); Dietrich VON LASSBERG, Dr.-Eng. and Peter HIRT, Dipl.-Eng., Augsburg (Chapter IV), and Fritz KUGEL, Dipl.-Eng., Heidenheim (Chapter V).

(*Eisenbahntechnische Rundschau*, November 1954.)

I. — PRELIMINARY OBSERVATIONS OF A GENERAL NATURE.

The first of a group of 13 Diesel locomotives was put into service on the 8th August 1953, by the « Ferrea Estrada de Leopoldina », at Rio de Janeiro, and the first of a group of 10 Diesel locomotives of the same design by the « Viaçâo Ferrea do Rio Grande do Sul », at Porto Alegre, on the 26th August 1953. Subsequently a further 21 of these locomotives has been added to the motive power park of these Companies.

What will interest the reader most, if he belong to a designing office, are the methods adopted by the builders to solve the problem put before them and the reasons upon which these methods are based. The opportunity of being able to make a comparative and critical examination of many new designs carried out at the same time all independently one of the other but all with the same object in view stimulates him in his own work.

For the designing Engineer, the accent is placed on « prudenter agas » but on the Operating Engineer on « respice finem ». The latter is only attracted by the published results if the locomotive which catches his favour has already proved itself in service. Whatever may be its economic drawbacks, he is not disposed to abandon the advantages of his good old steam locomotive which got his trains to destination in a reliable way even with leaking stays and loose big ends.

The authors of the present report do not pretend that the results obtained so far are other than provisional. Nonetheless, the hard conditions under which they were got, does justify a relative optimism.

These are the reasons why it was decided not to reveal till now a new design developed in secrecy. In the Spring of 1952, the « Maschinenfabrik Esslingen » received as a locomotive builder the order to design and build a first batch of three high power Diesel electric locomotives for the metric gauge in close collaboration with the « Maschinenfabrik Augsburg-Nürnberg », Diesel engine manufacturer and the « Maschinenfabrik J. M. VOITH », builders of the transmission. The order subsequently was increased to 23 locomotives.

II. — DETAILS OF THE PROBLEM AND THE PRINCIPLES ON WHICH THE DESIGN WAS BASED.

The locomotives were to be suitable for all classes of service, whether to

haul heavy goods trains or operate express trains at a maximum of 80 km (50 miles)/h. They had to be able to haul without assistance up the heavy grade of Santa Maria-Pinhal trains of about 300 t at about 20 km (12.5 miles)/h. This line has almost from one end to

Fig. 1. — 1 900 HP type C′C′ Diesel locomotive with hydraulic transmission.

the other up gradients of 36 $^0/_{00}$ with many sharp curves. The locomotives had to comply with the following specifications:

Maximum axle load	13.5 t
Minimum radius of curve . . .	70 m
Diameter of wheels	1 016 mm

Compressed air brake and vacuum brake for the train.

Control for multiple unit working.

Simplest possible driving controls.

Free running, in other words, suitable for operating over tracks of indifferent quality.

The climatic conditions given were: ambiant temperature 38° C, altitude 900 m (2 962 ft.) above sea level and humidity 100 %.

At the time the designs were taken in hand, there were no previous examples in modern construction of Diesel hydraulic locomotives of high power in Germany except the 1 B — B 1 Krupp of the Norwegian State Railways placed in service in 1940 and the V 80 locomotive of the Deutsche Bundesbahn, which was just completing its trials. The former was a rigid frame locomotive with two relatively slow speed engines, the second a bogie locomotive with a high speed Diesel engine.

The sharp curves and the ability to run over poor track led to the adoption of the bogie arrangement, the low axle load and the need for a substantial adhesive weight automatically called for the C′C′ layout.

To drive the locomotive two four stroke eight cylinder medium speed engines made by Messrs. MAN were selected. These series WV 22/30 A engines are supercharged and the supercharged air is cooled. The motor develops 950 HP at

900 r.p.m. under the conditions at Augsburg. To meet the climatic conditions given above, the power it limited to 850 HP at 900 r.p.m. The considerations, which led to the choice of the motors and their type will be dealt with by the maker himself.

Moreover, from both the technical and economic points of view it would have been difficult to carry it out. The results of the very valuable work done by the German Bundesbahn on the apparent moments occurring in multiple drive made it possible to justify the final deci-

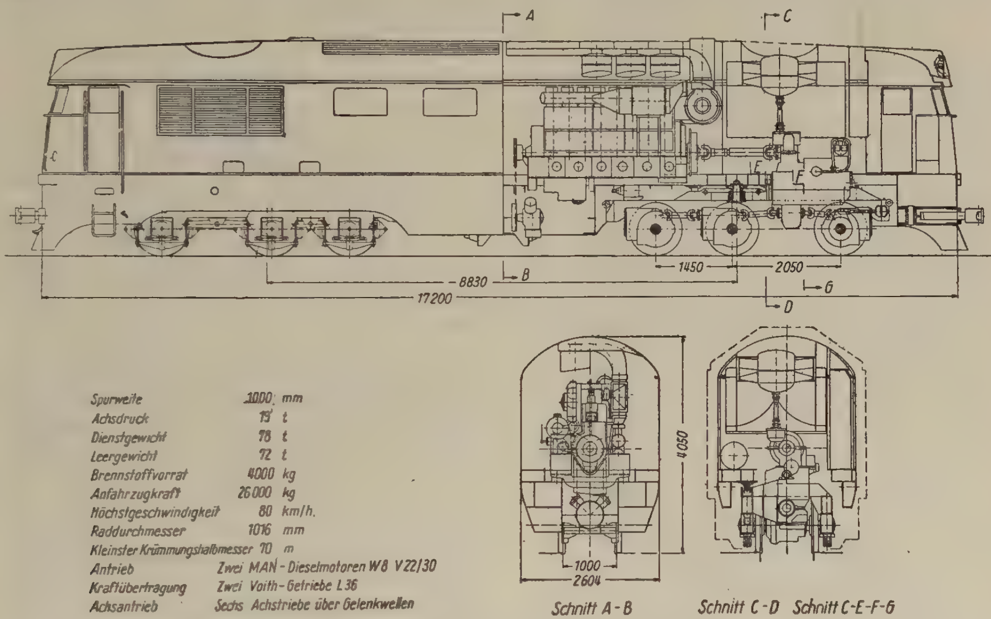

Fig. 2. — Longitudinal and cross sections of the locomotive.

Gauge	1 000 mm	Wheel diameter	1 016 mm
Axle load	13 t	Minimum radius of curve	70 m (229′)
Weight in running order	78 t	Power equipment: 2 MAN W 8 V 22/30 Dilsel engines.	
Tare weight	72 t	Transmission: 2 VOITH type L 36 hydraulic gears.	
Weight of fuel carried	4 000 kg	Axle drive: 6 axle drives with cardan shafts.	
Tractive effort at starting	26 000 kg		
Maximum speed	80 km (50 miles)/h		

When the design was started there was much controversy in the designing offices of all the locomotive builders about the relative advantages of individual axle drive and driving a group of several axles. A thorough investigation into the problem showed that the undoubted advantages of the individual axle drive, which makes it possible to bypass many difficulties, would only be obtained at the cost of a considerable additional capital investment and with certain risks.

sion to adopt this type of drive, provided proof were given of the necessary care in the kinematic layout of the cardan shafts and in their dimensions and in those of the axle drives having been taken. This decision also led to the selection of the VOITH L 36 r transmission with three converters with built in reverse gear driving through cardan shafts. The manufacturer of the transmission will give particulars of it.

The principles of the scheme were

established as follows: drive by two Diesel engines, each driving the three axles of one bogie coupled by cardan shafts through a VOITH hydraulic transmission.

III. — LAYOUT OF THE LOCOMOTIVE.

1. General arrangement.

A number of schemes were made and led to the arrangement shown in figure 2. The feature of this design is the placing of the Diesel engines in the body and the drives in the bogies. This solution is particularly satisfactory because:

1) the installation of the heavy Diesel engines near the centre of the locomotive gives a good moment of inertia about the vertical axis;

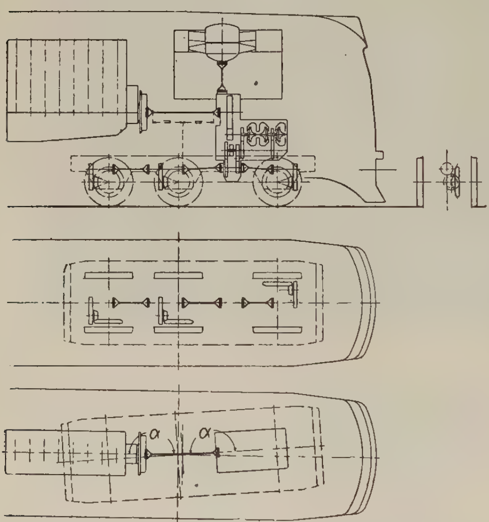

Fig. 3. — Arrangement of the transmission.

2) the installation of the water cooling equipment, calculated to meet tropical conditions, over the transmissions enabled a simple mechanical drive of the fans to be fitted without interfering with freedom of movement about the locomotive, accessibility, and ease of taking the equipment down;

3) the layout realised was neat and there were only unsignificant defects in the kinematics of the cardan shafts.

2. Transmission.

The system of shafts transmitting the power to the driving axles is subjected in power bogie locomotives to a series of movements in various planes. To observe the condition that the kinematics shall be exactly right is made difficult consequently and especially when it is necessary not to fit intermediate mechanisms.

The arrangement adopted (fig. 3) distributes these motions between two lines of independent shafts. The *horizontal* movements due to the displacement of the bogie and the relatively small *vertical* movements due to the action of the springs between the body and the bogie should be absorbed by the primary cardan shaft between the motor and the gearbox, as its speed of rotation is relatively slow. The symmetrical layout about the bogie pivot of the *engine* driven cardan joint and the driven cardan joint from the *gearbox* ensures the cardan shafts maintain an exact M form under all horizontal displacements of the bogie. In the vertical direction, should the springs deflect, the movement is always parallel in plane of the joints on the engine side and on the transmission side. The movement therefore is in an exact Z for the cardan drive, thanks to the elastic parallel guiding between the body and the bogies to be described later. At the maximum amplitudes, the horizontal and vertical angles of the cardans are well below the admissible limits (fig. 4).

The equality of the angles of the centre lines of the cardan shafts at all amplitudes ensures the transmission will have a constantly uniform movement of rotation at the primary shaft of the hydraulic transmission.

The short cardan shafts, revolving at high speed, driving the axle drives only have to absorb the vertical movements due to the play of the springs. They are not further affected by the movements due to the pivoting of the bogie.

The conditions specified of being suitable for running over lines of medium standard brought out the desirability of as large a vertical movement of the axles as possible. By the arrangement for balancing the reaction due to the couple being given the guided form shown in figure 5, it became possible to get the axle drives into a vertical position prac-

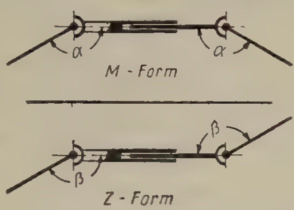

Fig. 4. — Cardan shafts arranged in M and in Z form.

tically parallel to the axleboxes guides and in consequence a movement of the driving universal joints in planes sufficiently parallel to one another and to those of the hydraulic transmission. In this way, a Z layout almost exact except for the small movements resulting from the chord of the arc described by the couple reaction balancing device even though the vertical movement of the axle can attain ± 45 mm (1 3/4 in.) was arrived at.

The method followed up to the present, and which consisted in absorbing the reaction of the couple by means of a vertical guide would, with the axlebox movement provided, have set up in the axle drive additional accelerations and decelerations. With this method of absorbing the couple reaction, each movement of the axlebox rotates the axle drive casing which motion is superimposed on the rotary movement of the axle, sets up additional inertia forces and can lead to undesirable torsional oscillations. Furthermore, the layout of the details for absorbing the couple reaction horizontally completely avoids the variation on the dead points and the disagreeable flexing of the springs set up when the locomotive starts.

The above indications refer in particular to considerations of the kinematics of cardan shaft drives and to steps taken in this matter. The layout of the locomotive transmission as a whole however gave rise to other problems which were solved by unusual means.

3. The method of carrying the bodyframe on the bogies.

It was not possible to arrange for the pivot to be a very low one on this metre gauge six wheeled bogie seeing that the axle drives were generously dimensioned and occupied almost the whole space between the wheel centres. The starting tractive effort of 16 t per axle applied to the lever arm of the pivot placed as low as the design allowed produced an appreciable turning moment and consequently a considerable unloading of the leading axle of the bogies and an addi-

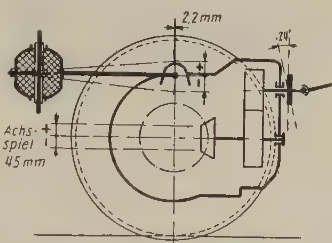

Fig. 5. — Arrangement for balancing the reaction of the torque.

N. B. — Achsspiel = axle play.

tional loading of the trailing pair. This unloading of the leading axle was thought to be undesirable from the point of view of safe running. Then too the variations in the axle loadings caused an unequal distribution of the power in the rigidly coupled transmission and a greater stress in the axle drives and cardan shafts.

The logical solution was to balance the bogie completely as between the three axles and to let the bogie then become unstable, bear against the underframe. With this

object a fourpoint bearing arrangement was adopted arranged in two vertical transverse planes. In this way, the pivot carries no load and only guides the bogie and transmits the tractive effort.

Pleasant riding of the vehicle which honestly can be insisted upon has to be obtained by the springing between the bogies and the bodyframe. Direct springing of the different bearing points is automatically excluded because it would not prevent the tilting of the bogies. As a result an elastic parallel guiding between the frame and the bogie was invented [1] and is shown in figure 6.

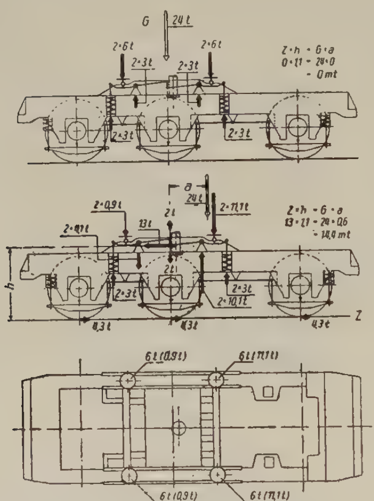

Fig. 6. — Parallel elastic guiding between the frame and the bogie.

The points of support of the frame on the bogie which are made in the form of sliding plates are arranged two by two on pairs of levers mounted longitudinally on the bogie frame. The levers of each pair are connected at one end by a guiding device and at the other rest on springs. The four points of support move exactly in parallel whatever be the distribution of the weight between them. To prevent the cup shaped bearings becoming inclined as a result of longitudinal and transversal play in the suspension they are carried in spherical bearings working in oil and air tight to avoid rusting of the rubbing faces.

When $Z = 0$ the four bearings are equally loaded. When $Z > 0$ the resultant of the bearing forces moves in the direction of running by the amount « *a* » and compensates the moment of the effort of traction $Z.h$. The uniform distribution of the load between the different axles therefore is unaltered even on poor track as the longitudinal compensation is perfect.

Furthermore, the elastic parallel guiding when the springs come into play ensures a vertical parallel displacement so important as regards the kinematics of cardan shaft transmissions between the motor carried on the frame and the transmission rigidly fitted to the bogie. The longitudinal balancing of the wheels and axles allows a considerable vertical movement of the axles even for an extremely small loading of the springs. In this way, it has been found possible to ensure the ability to run over bad track.

In practice it has been proved that the locomotive will take differences in level of the track of ± 45 mm without difficulty. The bogie rides very sweetly and this eliminates all undesirable effects on the hydraulic transmission which is rigidly attached to the bogie at three points. This soft riding has been demonstrated moreover by experience and involuntarily during several derailments on defective track which the enginemen had not noticed until after they had travelled even as far as 400 m (437 yards). The distance run on the sleepers had caused no damage. Figure 7 shows the investigation into the extent of the displacements of the axles on a shunting hump of 250 m (820 ft.) radius. Figures 8 and 9 shows the relative displacements of the

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(1) German patent No. 913 902, Class 20 d, Group 2.

axles of the bogie frames and of the body frame. The condition of the track on which the data was recorded is shown in figure 8. The curves of displacement of figure 9 were recorded on sensitive plates by lights carried on the details of the locomotive concerned.

To complete the list of problems in connection with the transmission, the coupling between the motor and the transmission must be mentioned. It consists of a combination of disc couplings with india rubber fittings and is also designed to slip. The function of the metal discs is to damp out vibrations. The slipping clutch portion is to hinder any propagation to the hydraulic transmission of peak couples occurring when passing through the critical speed of rotation when starting and when stopping the Diesel engine.

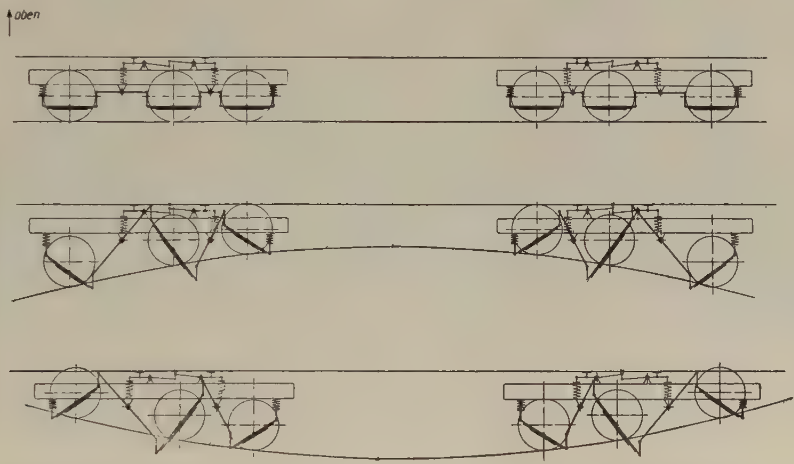


Fig. 7. — Motion of the wheels and axles of the bogies when running over a shunting hump

4. Bogie frame.

The bogie frame is fabricated by welding. The longitudinal members are of box section. The load of the body is taken by these side members, the body carrying supports, or the pairs of levers on which they rest, the roller bearing axleboxes, the suspension springs and the equalisers are all arranged on the vertical plane through the centre lines of these longitudinals. This simplifies the diagram of forces by eliminating all transversal moments. The pivot is located in the large middle cross stay. It is guided in the usual spherical bearing and can move vertically. Figure 10 shows the bare bogie frame and figure 11 the complete bogie.

The frame of the bogie was subjected to a very thorough investigation. The horizontal equalisers of the couple reaction produce considerable additional loads at points where usually there is no loading. By careful combination of the fundamental stresses due to the weight, the transverse shocks, the longitudinal forces due to braking, the guiding forces at the rails as well as those due to hauling the load, which were studied separately, the whole of the cases of loading were determined. In this manner, it became possible to calculate the stresses under conditions close to the actual ones, and so make the best possible use of the materials. The type of box section girder adopted has

been found very satisfactory from the mechanical point of view and to have great resistance to torsion. Some derailments due to the track have confirmed the correctness of the theory followed as regards the load and the mechanical layout.

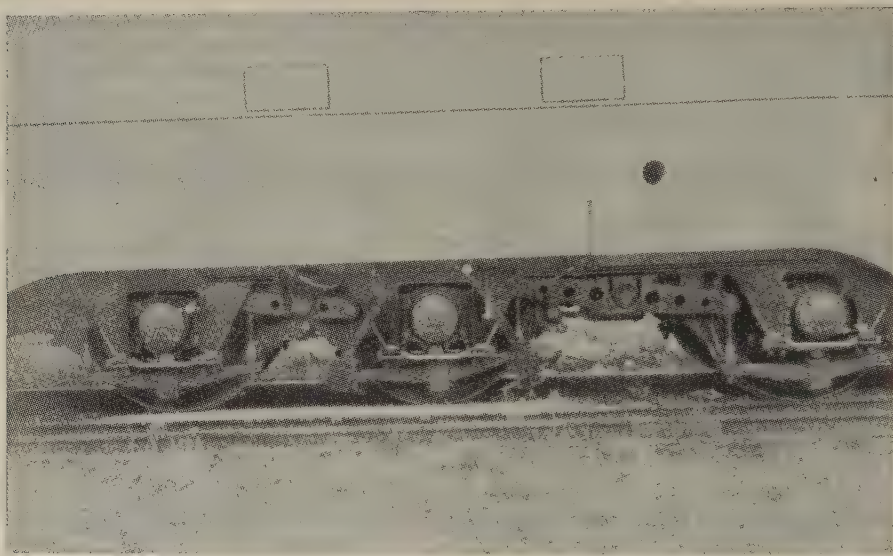


Fig. 8. — Passage of the locomotive over a 50 mm (2″) difference in level in the track.

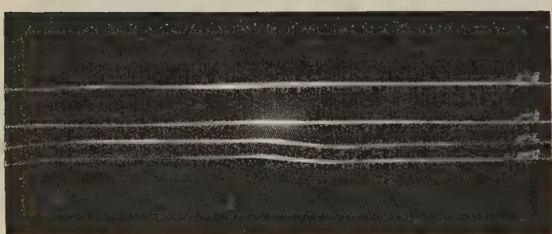


Fig. 9. — Characteristics of the movements in the case of figure 8.

a) the frame; *b*) bogie frame; *c*) rear axle; *d*) leading axle.

5. Locomotive body.

The body consists of the frame of light weight construction, completely welded, to which is riveted the superstructure (fig. 12). From the point of view of statics, it should be considered as a monoblock unit. With its two continuous longitudinals the frame forms the back bone of the body. The couplings with central buffers pivot on the headstocks connecting together the longitudinals. When designing the longitudinals provision was made for installing the piping and introducing in the carrying members troughing for the continuous pipe runs. This arrangement was most satisfactory both as regards construction and in operation. The fuel tanks to hold 4 000 l (880 gallons) have been built into the frame as well.

Relatively spacious driving compartments have been arranged at each end of the body. Next to these are the compartments containing the cooling equipment. In spite of the relatively narrow width of only 2 600 mm, it has been possible to provide adequate corridors and

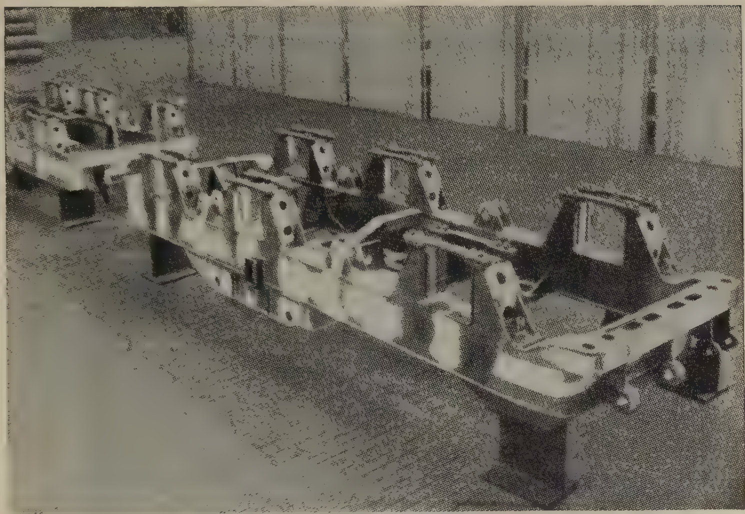


Fig. 10. — Bogie frame from below.

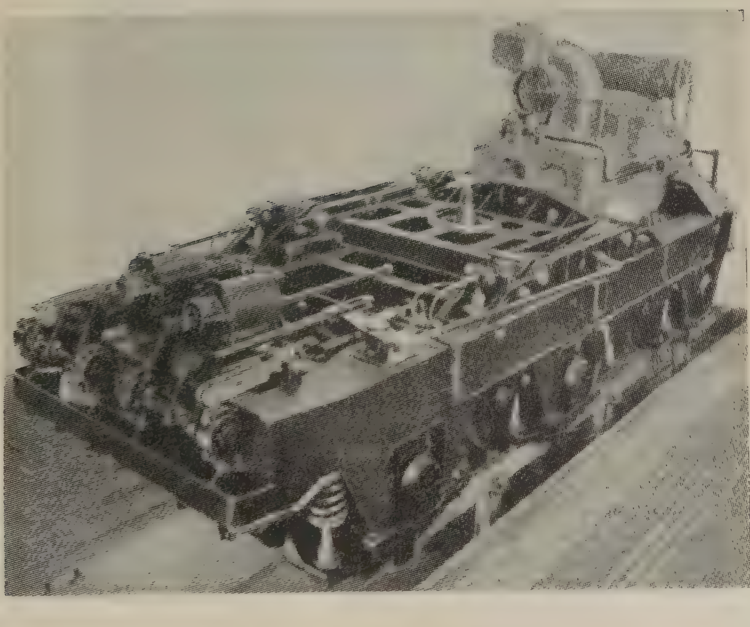


Fig. 11. — Bogie as a whole.

to make all the different groups of machine quite accessible. Figure 13 is a view of the engine room.

The sections of roof over the Diesel engines and the radiators are removable. Those over the engines carry the air ducts and oil filters for the combustion air which form an organic part of the roof. On lifting away the corresponding roof sections, the engine group, the cooler group, the hydraulic transmission unit can be lifted out of the locomotive.

The special characteristic of the locomotive body, considered statically, lies in the side walls and the roof playing their part in absorbing the stresses. This arrangement effects appreciable saving in weight. The main load to which the body is subjected is the buffing one. When calculating the body this load was taken as 200 t applied on the centre axis, 750 mm (2′ 5 ½″) above rail level. The tare of the body itself and of the machinery installed in it, only plays a secondary role : it is carried without any trouble. The distance between the particularly low centre line of the drawgear and the neutral axis gives rise to a bending moment of about 100 tm which produces a convex elastic line turned upwards. The weight of the equipment, carried in the body, acts in the opposite direction, and tends to reduce the deflection. The compressive stresses in the lower flanges of the frame under a buffing load of 200 t are of the order of 1 100 kg/cm². Every care was taken to keep well below the critical buckling stresses and to avoid

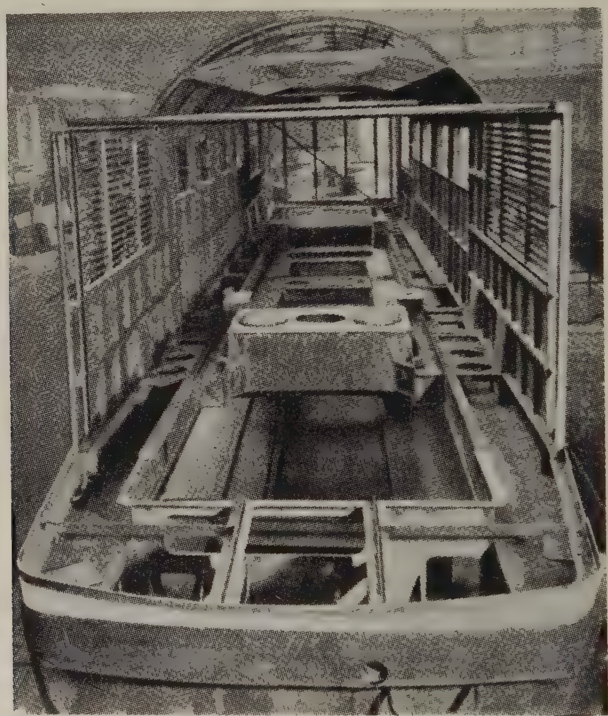


Fig. 12. — Body under erection.

local flexing. The box section longitudinals are of the same section their full length including the part occupied by the fuel tanks. As a consequence of this it was not necessary to include any means for changing the line of action of the buffing forces. This design also avoided any local sagging or extension of the frame.

The body was also designed to resist the twisting loads it can be subjected to in the event of derailment.

In order to check and confirm the results of the static calculations the locomotive was subjected to a test under the buffing load and a test under twisting. Figure 14 shows how the buffing load of 200 t was applied. In the foreground, the cylinder of a hydraulic press is shown inside the compression frame. At the opposite end a hydraulic measuring device is fitted and indicates the pressure on a pressure gauge which has been checked against a standard gauge. At about 40 points the elongations have been measured by wire resistances and transmitted to the electric measuring bridge seen in front. At the same time the deflections were recorded in order to be able to draw the elastic line of deflection. From the displacements of the panels of the side walls, like in form to a parallelogram,

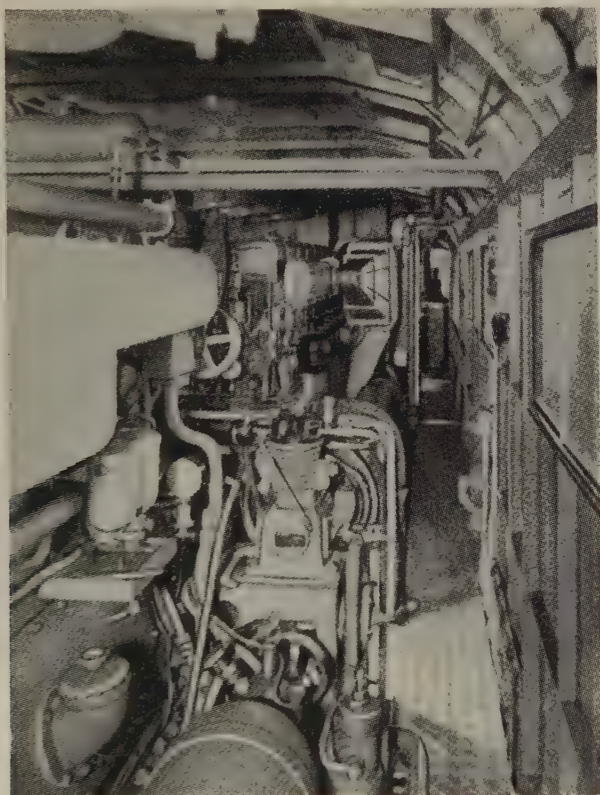


Fig. 13. — Engine room.

the proportion of the loads transmitted upwards to the roof by the pillars placed between the panels was calculated. Bands placed crossways on the panels of the side walls indicate the positions at which the displacements in the direction of the diagonals were measured by means of stretched wires and measuring devices. Bars placed transversely across the top of the body were used when measuring the twist. They were fitted at the ends with vertical straight edges, which were set by means of a level. At the back will be seen the lever which was loaded to produce the torsion moment. For this test, the body of the locomotive was supported at three points: one of these points of support is on the centre of the main cross stay whilst the other two hold the body solidly inside the second main cross stay. The weights of the details not fitted when making the test i.e. the engines, cooling equipment, fuel, etc., were applied at the correct places by means of a screw device with dynamometers. The supplementary forces due to the increase in flexion were eliminated by reducing the load from the screws.

Thanks to the measurements of the three components needed to ascertain the behaviour of the body as regards mechanical strength: elongations, deflections and displacements, the tests under load gave a clear picture of the static behaviour of the locomotive body. By taking into account these three phenomena of deformation differing from each other but interdependent from a static point of view it has been possible to establish an equation between the values measured and so form an appropriate statical picture.

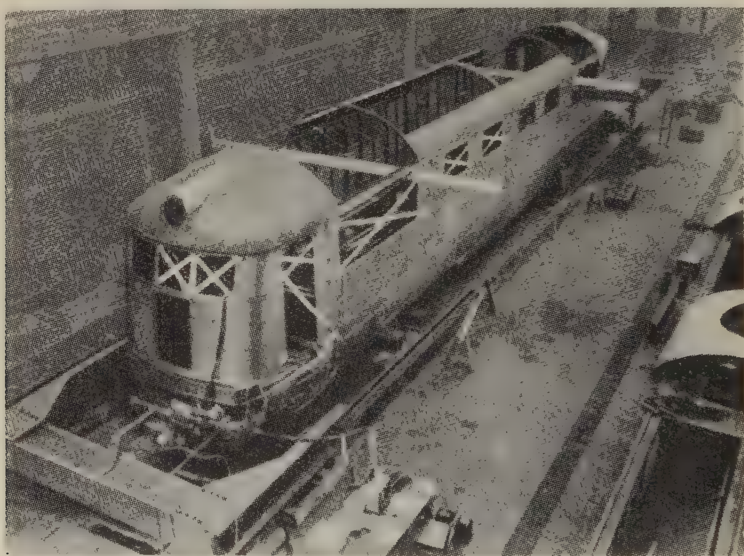


Fig. 14. — Test under buffing pressure.

6. Installation of the engine.

In the locomotive the most appropriate bed would be the frame itself, but with the lightened designs inevitable with rigid limitations of weight, elastic deformations of the frame are to be expected, which as experience has shown would have

undesirable repercussions on the Diesel engine crank case itself.

These considerations led to the engines being carried on sturdy sub-frames which are supported by the main frame through metal bearers with rubber pads (fig. 15). The sub-frame has not only to carry the weight but also to help the engine crank case to support its own internal forces. As regards the bearers, they have to deal with the elastic deformations and prevent them from affecting the Diesel engine crank case.

Each engine bed carries an air compressor for the brake, an air compressor for starting, a dynamo for lighting, the two compressed air bottles for starting, the lubricating oil cooler, and the brake compressor intermediate radiator. In addition, the sub-frame carries the lubricating oil filter motor and the lubricating oil feed pump. In this way, a completely self contained unit was achieved which could be put into the locomotive fully erected with all its piping. This set is carried on the main frame at four points. Horizontally, the set is secured by four rubber lined bushes threaded on bolts locked into the bearings. The two bushes on the side opposite to the couplings have a free space in the direction of the longitudinal axis which makes it easy to move the unit longitudinally. The transversal fastening device includes a rubber spring.

The installation of the engine, auxiliaries and piping, which can be done in spite of the exiguity of the locomotive compartments, materially reduce the work in the shops. The replacement of a complete engine set, including all ancillary work, takes about four working hours. The engine cooling water pipes are connected to the cooling installation and the compressor to the brake piping by rubber pipes.

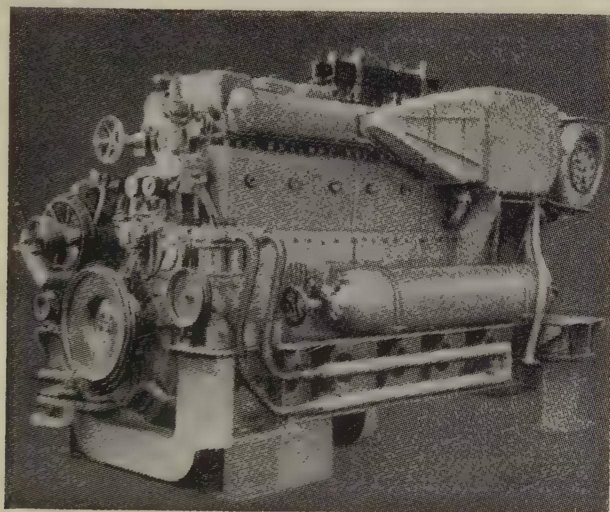


Fig. 15. — Engine frame.

7. Radiator installations.

The cooling of the engine water, engine lubricating oil, and the oil of the hydraulic transmission is looked after by VOITH coolers. Each engine transmission set has one of these coolers. The lubricating

oil and the transmission oil are cooled indirectly by heat exchangers. Each cooler set is divided into two circuits. The two cooling water pumps however are combined and fitted immediately alongside the engine. The main circuit includes the engine radiator and the transmission oil heat exchanger : the secondary circuit includes the radiator for the super charging air and the heat exchanger for the lubricating oil. The cooling set gets its air supply from an axial fan driven through a cardan shaft and a VOITH hydraulic regulating coupling, by a bevel-gear on the speed increasing gear of the transmission. The speed of rotation of the fan is regulated by varying the filling of the coupling, the degree of filling being controlled thermostatically as a function of the cooling water temperature.

The cooling equipment is the result of very thorough investigation both at the VOITH aero-technical testing plant at the VOITH Works and — after erecting — on the locomotive test plant in the Esslingen Works (2).

8. Brake equipment.

The locomotive is equipped with the Westinghouse automatic continuous air brake with a separate vacuum brake equipment for braking the train. The trains can be braked by compressed air brakes equally as with vacuum brakes. The locomotive is fitted only with the compressed air brake which is applied through a type VD valve when the vacuum brake driver's valve is operated. The brakes act on both sides of every wheel. The maximum brake percentage is 65. A dead man's device assures the train stopping if the driver becomes a casualty. When running double headed, the supplementary brakes are applied on each locomotive. In this case too the compressors and exhausters run in parallel which doubles the braking capacity. Thanks to this arrangement in combination with multiple working to be dealt with later, double heading is not limited to occasional assisting short trains up steep gradients but can also be used to work heavy loads that is trains of considerable length.

9. Control.

To operate several independent motor units from a single given control cabin, as in double heading, requires the multiple control gear to be thoroughly reliable. Preference was given to compressed air control over electric control as being simpler and more reliable. The details of the compressed air control, which are similar to the details of the compressed air brake, as regards upkeep and inspection do not require the specialist knowledge needed with electric control by means of relays, which the drivers lack.

Compressed air control developed in conjunction with the Hanover Works of the Westinghouse Company and incorporating many standard details is supplied with air from the main reservoirs through a pressure reducing valve. The main detail of control the « manipulator » is formed by a combination valve with a rotary valve giving fine regulation which fills the hydraulic transmission and speeds up the motors. The compressed air cylinders needed to regulate the speed of rotation of the motors and to regulate the filling device of the hydraulic transmission are built into this group.

The manipulator has the following settings :

A) *Neutral position :*

Used when running double headed and when the driving compartment is not occupied and allows the manipulator handle to be removed.

(2) The cooler installation and the test results appeared in a separate report in *Glasers Annalen*, No. 6, 1954.

I. *Position when running light :*

The engine runs at is idling speed and the hydraulic transmission is empty.

II. *Filling position :*

The motor continues to run at its idling speed and the hydraulic transmission refills.

distribution valve. This valve can be operated only if :

a) the motor runs at its idling speed;

b) the hydraulic transmission circuits are empty;

c) the locomotive is stationary.

A control valve fitted to one of the axles of the locomotive only releases the

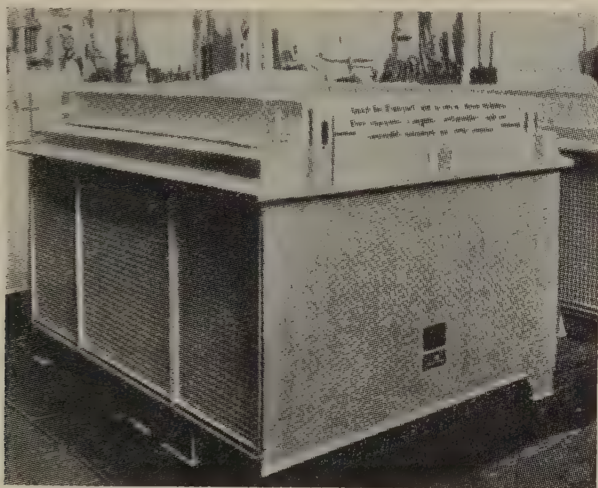


Fig. 16. — Cooler set.

III. *Running position :*

Moving the manipulator from the filling position causes, through the operation of the fine control valve, the speed of the engine to increase up to its maximum. The two Diesel engines and hydraulic transmissions are coupled in parallel. In figure 17 is shown the continuous variation of the speed of rotation of the two engine sets of the locomotive in terms of the control pressure obtained by operating the manipulator. The satisfactory agreement of the two engine sets will be noted.

To change the direction of running each driving compartment has a reverse control consisting of a compressed air control of the reverse gear when the axle ceases to revolve. This valve was designed by the locomotive builder — using with his agreement — an idea of Erich Burmeister and was made by the Westinghouse Company. It is usually known in the Esslingen Works as the « Burmeister valve ».

This valve (fig. 18) consists of a small cylinder supplied with compressed air through a pilot valve at the driver's desk. The piston of this control appliance is fitted with an articulated spindle which bears on the axle which is intended to control the stop, or against any other shaft whose rotation depends upon the locomotive being in motion. If the shaft is still revolving, the articulation of the

spindle acts and the piston uncovers the exhaust ports so that the pressure cannot rise in the feed pipe of the gear. Consequently the interlocking of the reverser connected to this feed pipe is not released. The reverser lever cannot be moved. It is only after the shaft in question is stationary that the articulated spindle ceases

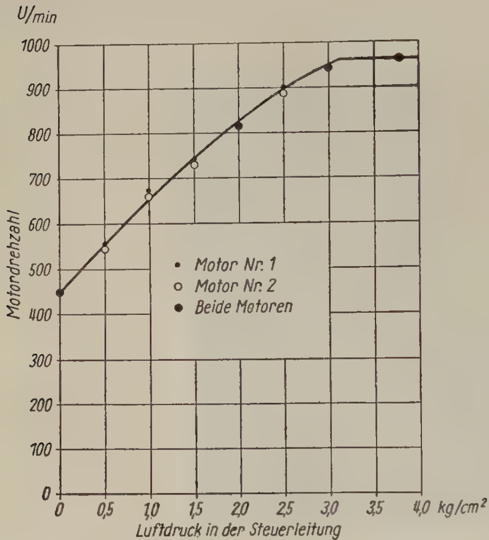
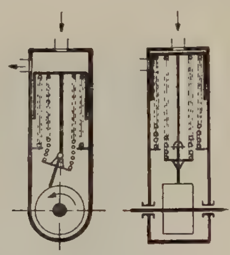


Fig. 17. — Relation between the speed of rotation and the control pressure for the two engines of one locomotive.

N. B. — Motordrehzahl = speed of rotation of the engine. — Luftdruck in der Steunerleitung = air pressure in the control piping. — Beide motoren = the two engines.

to be displaced and the exhaust air ports remain closed. The pressure which then builds up unlocks the reverser.

When reversing, the manipulator remains locked in position I (running light) and is only released after the reversal of all the transmissions (when running double headed, four) has been checked. If one of the two — or of the four dog-clutchs — remains in the tooth on tooth position, it is impossible to increase the speed of the engine whatever be the position of the manipulator. The speed of the engines can be freely regulated only when the dog-clutchs of all the transmissions are in the end position. Furthermore, the position of all the reverse dog-clutchs is shown to the driver by two indicator lights of which only one lights up in the position tooth on tooth, whereas both light up in the final position. Thanks to having adopted the



Running. Stopped.

Fig. 18. — Burmeister valve.

series coupling for the control gear a perfect control of the reverse is obtained very simply with these two easily seen indicators. Figure 19 shows the layout of the operating gear and the indicating gauges in the driving compartment.

10. Electrical installation.

For the reasons already given for selecting pneumatic control, the electrical installation of the locomotive was kept as simple as possible. The current is produced by two lighting dynamos each driven by one of the Diesel engines. The electrical energy is stored in a battery of accumulators. The electrical installation is primarily intended for lighting the locomotive. As there are two driving compartments electric gauges are fitted to show how the two groups are working. To transfer the fuel from the lower tanks to the raised auxiliary tank an electrically driven pump is provided. The control of the operation of reversing the locomotive and the locking of the Diesel engine speed when the reverser has not

completed its movement is also ensured electrically. Figure 20 illustrates the amount of electric equipment installed.

IV. — INSTALLATION OF THE DIESEL ENGINES.

1. Choice of Engine.

To drive the locomotive two 4 stroke 8 cylinder Diesel engines made by M.A.N. at their Augsburg Works were chosen. These are engines of the WV 22/30 A type with precombustion chambers and are supercharged by exhaust gas turbo-supercharger (fig. 21 and 22). Under the conditions at Augsburg each engine developed 950 HP at 900 r.p.m. The diameter of the piston is 220 mm and the stroke 300 mm and the mean piston speed is 9 m/sec. The mean effective pressure 10.2 kg/cm² corresponding to the relatively high supercharging of 85 % was made possible by cooling the supercharging air. The engine as installed has a specific weight of 6.3 kg/HP.

The above dimensions of the engine and its speed of 900 r.p.m. show that it is not a typical high speed engine. Such an engine could equally well have been employed when designing the locomotive, the more so as such engines are mass produced by M. A. N. at Augsburg. A

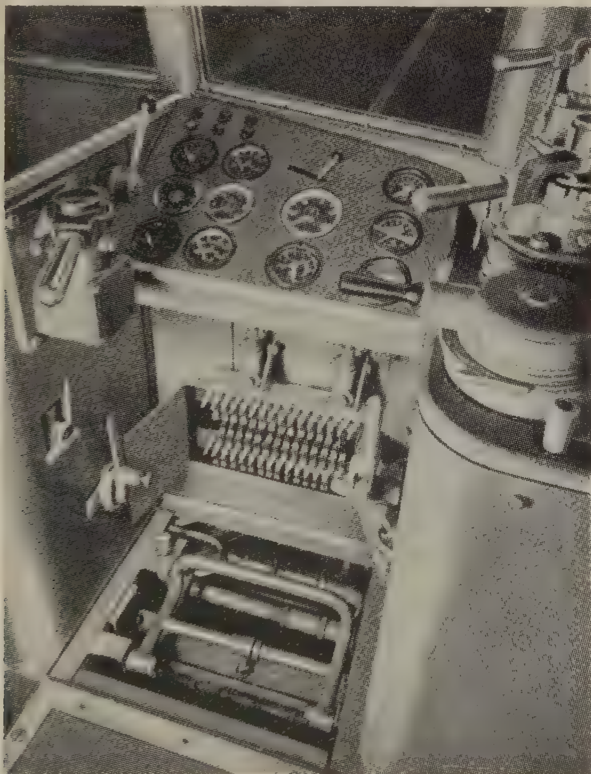


Fig. 19. — Layout of driving cab.

12 cylinder four stroke Diesel engine running at 1 500 r.p.m. instead of 900 r.p.m. would have developed the same horsepower and thanks to its higher speed would have weighed only 4 kg/HP. This lighter engine would have reduced the total weight of the locomotive by 4 t or by only 5 %. On the other hand, the use of a medium speed engine had a number of advantages. Whereas high speed engines are usually fitted to the bogies of railcars and their dimensions are relatively small, more robust equipment is preferred for locomotives. The lower speed of rotation, the fewer cylinders, and the improved accessibility to all parts of the mechanisms are valuable features both in operation and as regards economy. A comparison with the many slow speed American Diesels shows nonetheless that the speed $n = 900$ of the type WV 22/30 A is still higher than that of most of the engines in American locomotives.

Another aspect of the problem favouring the use of a relatively slow speed engine and therefore a more robust one was the difficulty of finding in the country in which the locomotives were to operate sufficient staff with the experience needed to maintain high speed engines. The thermic stresses and the fuel consumption are lower with slow speed engines. The distances run consequently are longer with locomotives fitted with slow speed engines than with those with high speed engines. These arguments weighed the balance in favour of the lower speed engine.

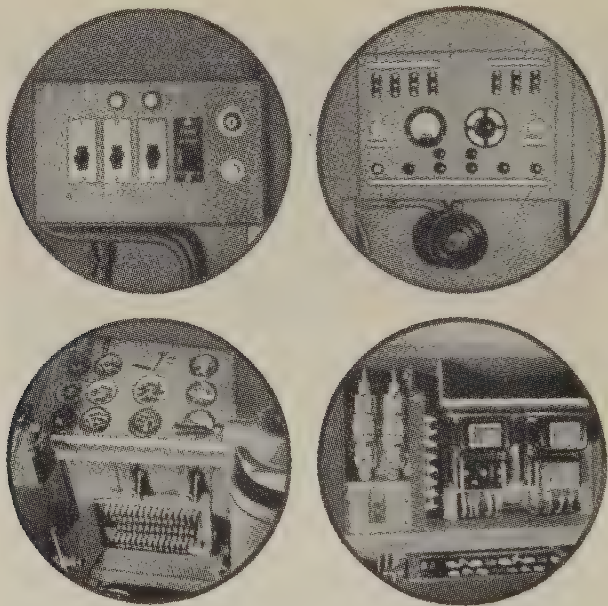


Fig. 20. — Electrical installation. *At the top :* distribution boards; *to the left :* in the driving cab; *to the right :* in the engine room; *at the bottom, to the left :* gauges; *to the right :* battery regulator.

2. Description of the engines.

The *crankcase* is of special cast iron with substantial ribs to stand up to the stresses set up on ignition. It can be secured directly to the engine sub-frame without any special foundation. The base is closed only by the light welded sheet steel sump which contains the engine lubricating oil. This layout (fig. 23) made it possible to build the engine with a very low specific weight for an engine running at a medium speed.

In this design the *crank shaft bearings* are supported from the crank case. The substantial steel bearing caps are bolted to the crank case by special quality steel bolts and are located transversely between fitted faces. The bearings are steel with a thin layer of lead bronze. The crank shaft bearing nearest to the coupling ensures the axial location of the crank shaft. The lubricating oil comes in from below through the bearing caps : the oil is however not supplied from below to the sliding surfaces of the bearings where the pressure on the bearings is highest but laterally so that the lower bearing subjected to the greatest loads does not break the continuity of the oil film. The *crank axle* is of special high quality steel. It runs in nime bearings and the crankpins are flame hardened. Thanks to this and the use of lead bronze bearings, the bearings can stand high specific loads. To reduce the weight of the revolving parts and the forces, the crank shaft is hollow, the bored holes being fitted covers and being used to convey the lubricating oil to the connecting rods. Counter balance weights secured to the four crank discs further reduce the inertia forces acting on the bearings.

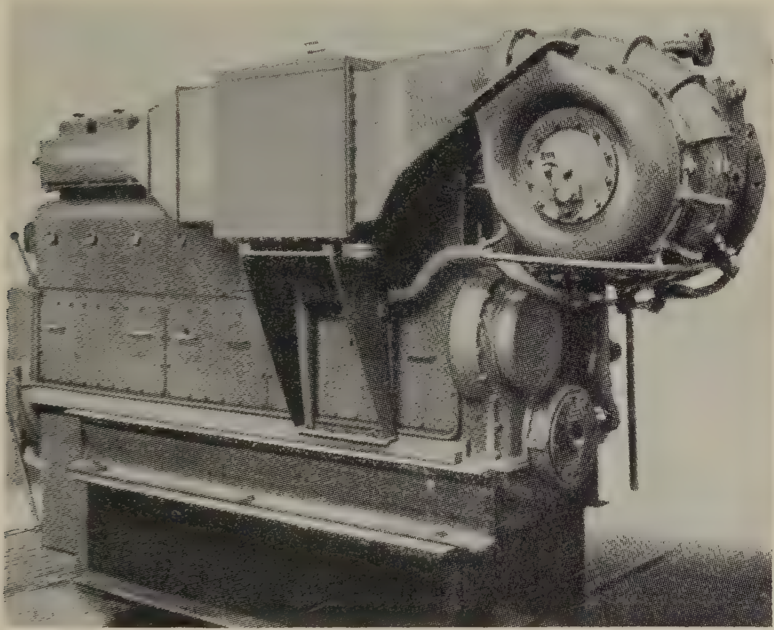


Fig. 21. — MAN W8V 22/30 A Diesel engine; valve gear side.

At the end opposite to the coupling

the crank shaft is fitted with an *oscillation damper* which works on the principle of the spring and sleeve and protects the shaft from dangerous torsional oscillations. As an extension in line with the crank shaft at this same end is a shaft connected to it by a flange which carries a *pulley for a trapezoidal belt drive,* to drive the compressor for the air brake, the compressor for the compressed air starter, the lighting dynamo and the vacuum exhauster.

The H section connecting rods are drop stampings. The big and little ends of the connecting rods have steel bearings with lead bronze linings, the lubricating oil being fed through the crank shaft and the hollow connecting rods.

The *pistons* are cast aluminium alloy which reduces the oscillating masses and especially owing to the high conductivity of the metal keeps the pistons and piston rings down to a low value in consequence of which perfect results are obtained even under continuous heavy loads. Thanks to the high conductivity of the metal and to their rational design the pistons were made without providing any cooling device. Four piston rings transmit the heat to the cylinder liner and seal the combustion chamber from the crank case. In addition three scraper rings remove the lubricating oil thrown up from the crank case from the cylinder liner walls.

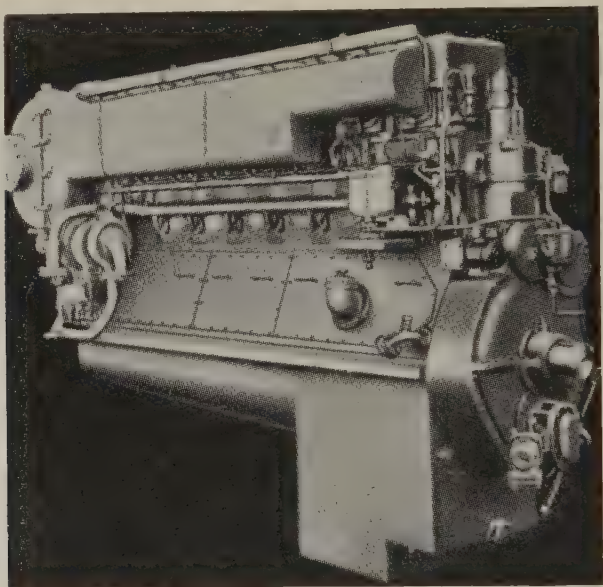


Fig. 22. — MAN W8V 22/30 A Diesel engine; exhaust side.

The *cylinder liners* are made from special cast iron of a particular structure which reduces wear to a very low value. The sleeve is ground into the cylinder block to ensure there be no leakage from the cooling water jacket. At the lower end, three rubber joints make the seal, a leakage groove being provided to let any water which might accumulate between the joints escape. Any discharge of water would indicate any possible

defect of tightness. The combustion chamber is made gas tight by a copper gasket between the liner and the cylinder head block.

The *cylinder heads* are secured by four bolts of special quality steel. They are fitted with two admission and two exhaust valves, a starter valve and a check valve, and have a central pre-combustion chamber. The top of the heads has a rim to collect any oil escaping from the lubrication points and allow it to be drained away. Each head has a cover which keeps it oil tight and damps out noise.

The engines are fitted with *Bosch prechamber and nozzles* which make them indifferent to variations in the quality of the fuel because the nozzles have no fine holes and clean themselves automatically.

The *cam shaft* revolves in white metal bearings in the engine block. It is in two pieces to make it easier to fit it from the leading end of the engine. The admission and exhaust cams are solid with the shaft, but the fuel cams are separate as they have to be adjusted to get equal combustion pressures and to ensure the fuel consumption is good. The cam shaft is hollow, oil for its bearings being pumped through it.

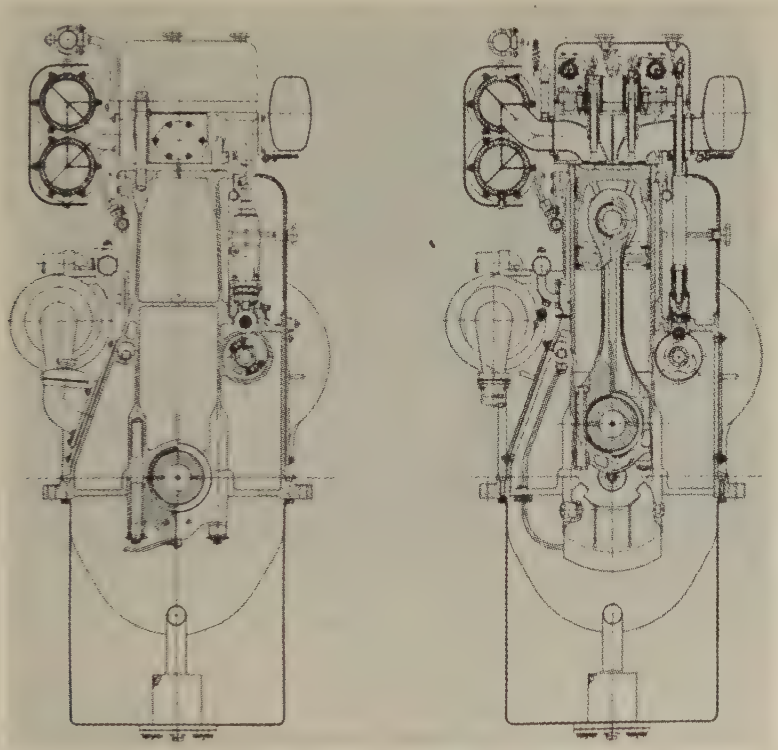


Fig. 23. — Cross section of the Diesel engine.

The *valves* are operated by push rods fitted with rollers, rocker rods and rockers. These rockers operate the valves through

rollers which relieve the valve guides from side thrust. The exhaust valves fitted on the opposite side of the engine to the valve rockers are driven through intermediate levers and an articulated connection. The whole of the valve operating gear is connected to the pressure lubricating system, and is enclosed to make it oil tight and quiet. The push rods are located in tubes into which the lubricating oil from the sleeves passes to ensure the rods are properly lubricated and then empties into the crank case.

Two inlet and two exhaust valves ensure the cylinders being efficiently filled. The valves are carried in guides fitted in the heads which are oiled by oil dropping from the rollers of the rockers.

The *fuel injection pumps* are independent Bosch pumps. The fuel from leaks is carefully kept separate from the lubricating oil and drained away separately.

The *regulator of the speed of rotation* is driven off the opposite end of the cam shaft to the coupling joint end. A spring fitted device protects the regulator from possible irregularity in working. To make it easier to operate the regulator, a servo-motor driven by oil under pressure is fitted. The regulator springs and thereby the speed of rotation are controlled hydro-pneumatically.

In addition to the regulator control, at the end of the engine opposite to the coupling, the other *auxiliary drives* are fitted. A toothed gear wheel fitted to the crank shaft drives the lubricating oil gear pump, which draws the oil from the lowest point of the crank case, forces it through a filter and a cooler before it is distributed to the various points to be oiled. The cam shaft drives the fuel feed

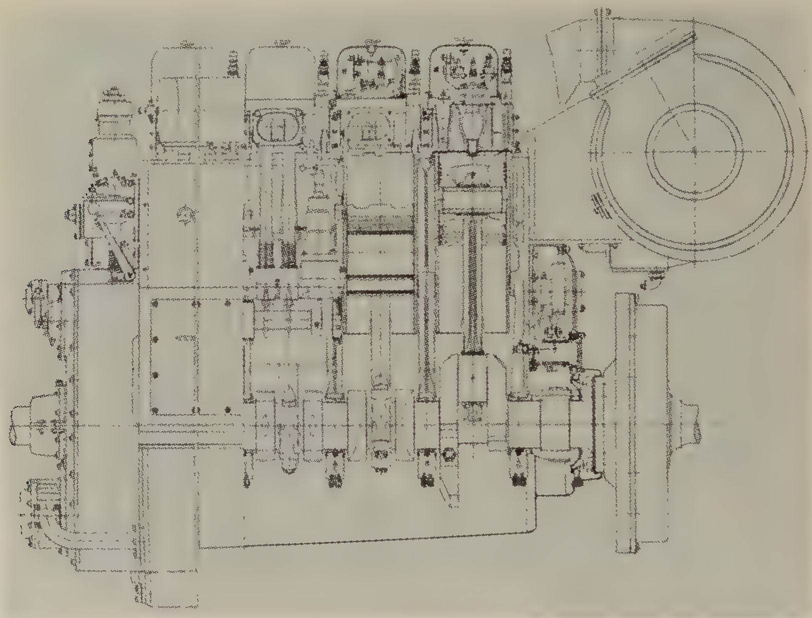


Fig. 24. — Longitudinal section of the Diesel engine.

pump, also a gear pump, which draws the gas oil from the reservoir and passes it through a filter to the fuel injection pumps. The electric tachometer generating mechanism is also fitted at this point.

The gears of the cam shaft, arranged beside the coupling, drive the two *cooling water centrifugal pumps*. The two are carried on the same shaft. The large pump discharges warmer water for the engine and transmission oil cooler, the smaller the colder water for the supercharger air cooler and the engine oil cooler. This division of the water circuits meets the two different conditions: whereas in order to reduce wear and in view of the smaller cooling surfaces it is desirable to run the engine at a relatively high temperature, the supercharger air radiator which cools the air heated in the compressor ought to be maintained at a lower temperature seeing that the temperature of the air controls the weight of air fed to the engine and thereby the power of the engine and the way it behaves in service.

The engine *is started* by compressed air by means of pneumatically operated valves grouped in star formation round the coupling end of the engine and controlled by one cam on the cam shaft.

At the same end of the engine the Brown Boveri *turbo-compressor* group is fitted (fig. 21). It includes a single stage exhaust gas turbine and a single stage radial turbine. The rotor is carried at each end in roller bearings. The maximum speed of the supercharger group is 15 000 r.p.m.

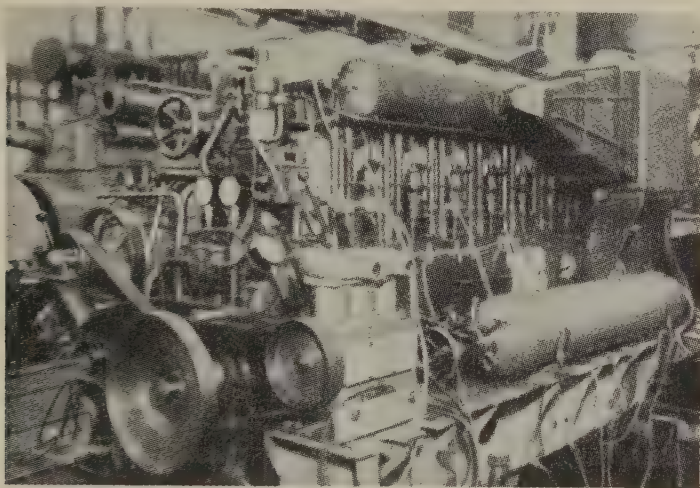


Fig. 25. — Power unit on the test plant.

The supercharged air and air cooler piping is arranged on the same side as the cam shaft. The *exhaust gas pipes* divided in accordance with the Büchi system are located on the opposite side of the engine (fig. 22).

3. The installation of the engine in the locomotive.

The normal engine with plain bearings and with side covers on the crankcase provides ready access to the various parts of the mechanism. It is not essential

to lift it out for repairs which could quite well be carried out with the engine in position in the frame. In the present case the usual layout was abandoned and the engines were carried on separate sub-frames through rubber fitted brackets because this layout was necessary for constructional reasons as explained in Chapter III. Apart from protecting the engines from outside stresses that could not be foreseen, this method of construction has other secondary advantages: the periods the locomotives are out of service for overhaul can be reduced to the minimum as a complete power unit can be replaced by another in working order in a few hours. Furthermore, the elastic supports isolate the locomotive frame from high frequency vibrations and this suppresses the noise due to resonance in the body of the locomotive.

The very restricted weight allowed made it necessary to adopt a lightened form of construction even for the engine carrying brackets. In order to check the value of the type of construction adopted, the complete engine installation including the elastic support was tested at the M. A. N. testing plant. Recording instruments did not show any disturbing resonance. Vibration measurements showed that the amplitudes of oscillation under operating conditions were of the order of magnitude of $\pm$ 0.2 to $\pm$ 0.4 mm, which agreed closely with the original calculations. These vibrations were considered as negligible in practical working.

The effect of any deformation, always possible, of the frame upon the engine supports and on the engine itself was the subject of an investigation under static conditions. The engine support resting on its rubber mounted brackets was tilted by 28 mm on an angle, and measurements taken at all the cranks webs of the crankshaft showed there was no change in the dimensions of the webs.

This showed that the flexibility of the elastic carrying brackets allied to the rigidity of the engine sub-frame prevented any undesirable stresses from affecting the engine.

In all these investigations extensiometric measurements were taken on the engine carrying brackets and on the engine itself. The stresses were found to be very low in all cases. Figure 25 shows the engine and its sub-frame when these measurements were being taken.

4. Coupling between engine and transmission.

In order to protect the shafts in the transmission gear from excessive stresses throughout the whole range of stresses in service the Diesel engine flywheel is fitted with a coupling of the type known as the frictional metallic disc type (Schwingmetall Rutschkupplung) (fig. 26). With this type of coupling the maximum angle of drive is 3°. The power is transmitted through two rubber pads vulcanised on steel discs. The couple is transmitted through the friction members to the rubber coupling. The friction coupling is set to transmit a couple 2 ½ times the average power to be transmitted and to slip above this value: this occurs during a short interval when passing through the critical speed of about 325 r.p.m. In practical operation critical speeds are not experienced.

When designing this coupling further other important factors had to be taken into account. The coupling had to be able to drive the cardan shaft even as occurs, when the bogie is pivoting on curves, if considerable transversal forces are to be handled. The secondary member of the coupling moves relatively to the primary one owing to the elasticity of the coupling and this ensures adequate and constant lubrication. Moreover, the rubber and friction parts of the coupling are adversely affected by mineral oils and greases. It was therefore necessary to provide suitable joints of « Buna ». If the lubrication is excessive, the excess can escape through the oblique hole

drilled in the flywheel casing. The coupling as made is very compact to meet the installation restrictions imposed and is very light. This result was due to the coupling being built into the flywheel of the Diesel engine. After removing the cardan shaft which can be moved back a little, the whole of the coupling can be taken down.

The couplings have met all demands on them both under test and in service.

5. Air filters.

When taking the design in hand, it was known that in practical working much dust was to be expected. The necessary air filters, able to hold the largest possible volume of dust of any existing commercial make, could not be located on the locomotive neither as a relatively large number of small filters nor as filters of normal dimensions. Here again a new investigation had to be taken in hand. Air filters relatively thin but of exceptionally high dust capacity which could be installed partly in the curvature of the roof were manufactured. Over the engines three oil filters of the Mann and Hummel type (fig. 27) were provided.

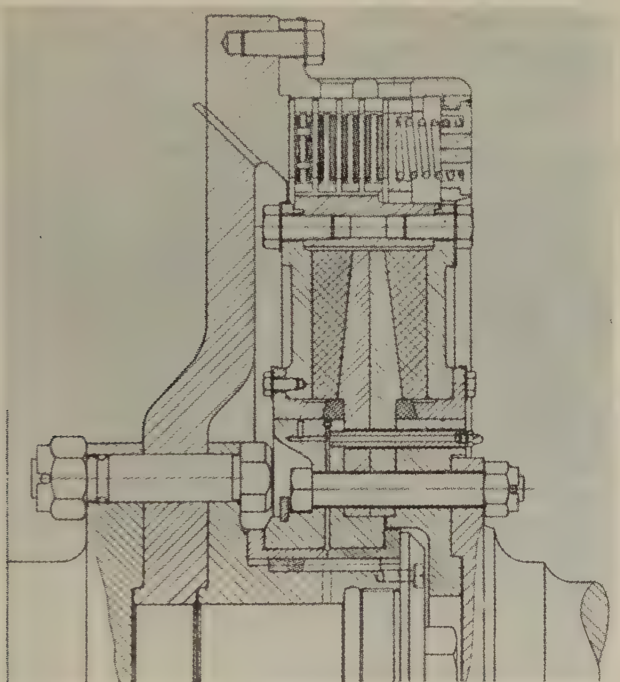


Fig. 26. — Elastic and sliding clutch.

These filters are simple to maintain. From time to time the oil with the dust entrapped in it is emptied through the dirty oil drain and replaced by new oil, which can be used oil from the engines. The internal fittings in the filters do not need cleaning, the fresh oil does this work. When the locomotive undergoes general repairs the lower collector of the filter is taken down when the last traces of dirt are cleaned away.

Oil bath filters are relatively flat: they have a stiff casing and are particularly suitable for the severe duty of locomotives. At the M. A. N. test plant their action was carefully measured and it was found they eliminated 98 % of the fine dust. The tests also showed that owing to their relatively low height the air resistance to be overcome in these oil filters was negligible: it reached 100 mm of water in the case of a clean filter and 130 mm for a dirty one.

6. Lubricating oil filters.

To filter the lubricating oil, combination filters of plates and gauze screens are used. The plate filter is the first and collects the coarsest impurities: the gap between the plates is 0.12 mm. The gauze filter is built up with gauze screens in the form of plates, the gauze having a mesh of about 0.1 mm. The arrangement of the filter and the path taken by the oil is shown in figure 28. The lubricating oil filter is carried on the engine bed between the heat exchanger and the engine and is quite accessible so that the driving and maintenance staff can rotate the plates and empty the dirt collected easily.

V. — THE TRANSMISSION.

1. Fundamental considerations on the transmission of the power.

As we have shown in the First Part, the locomotives had to be general purposes locomotives to be used as much as possible for all services. They had to be able to haul heavy goods trains up long severe gradients equally as well as lighter passenger trains at high speeds. A very thorough investigation into the different methods by which the power could be transmitted to the wheels was undertaken in order to arrive at the solution which would meet these very wide requirements with good efficiency whilst at the same time giving reliability in service and at the lowest cost.

It was soon realised that to transmit the high tractive effort needed with an axle load limited to only 13 t would involve the use of six driving axles. A further question was should the locomotive have individual axle drive, i.e. should each pair of wheels be driven by a special hydraulic drive including reverse gear or if all the axles or groups of axles coupled together should be driven by one set of hydraulic drive. This latter

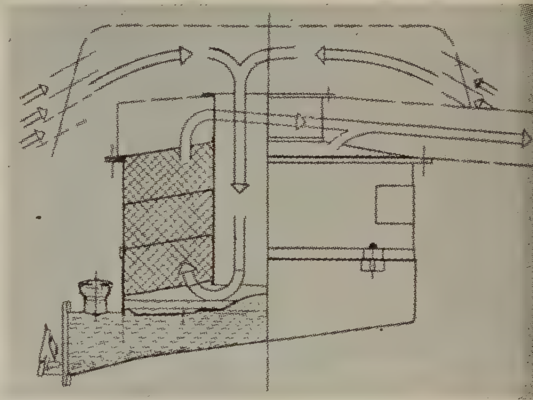


Fig. 27. — Oil bath air filter (Mann type).

drive is not only cheaper to build but its operation and the control of the transmission is simpler and clearer. It also has the advantage of reducing the likelihood of slipping of the wheels. At that date when the decision had to be made, and upon which the arrangement of the locomotive largely depended, the fundamental knowledge both theoretical and practical needed was still far from complete.

Objections without end were raised against the rigid coupling of a number of axles through toothed gear and cardan shafts. Stress was laid on the unavoidable differences between the driving wheel diameters and as a consequence on

the considerable resistance and reaction which might reduce the useful life of these parts. To meet these objections a considerable number of railcar bogies were fitted with independent drive of the VOITH type with double transmission each half of which drove simultaneously the two axles at low speeds and high tractive effort or alternatively one axle at high speeds. The results obtained were excellent. These tests had also to be considered with the results obtained with innumerable steam locomotives with a number of axles coupled by rods and which run without difficulty. The good results obtained with bogie railcars with axles driven and rigidly coupled by the cardan shaft and bevel gear axle drives and with which the expected difficulties did not arise were also known.

The difficulty was due to the generally accepted idea according to which when for example two wheels of different diameter rigidly coupled together rolled freely as in figure 29 (top left), there should be slipping i.e. friction of the wheels relatively to the rails and this could only occur by overcoming the known high coefficient of adhesion as shown by the full line curve of figure 29 (below) as a function of the running speed (coefficient of adhesion according to Kother), which gives especially at high speed the powers absorbed. These are several times greater than the available power.

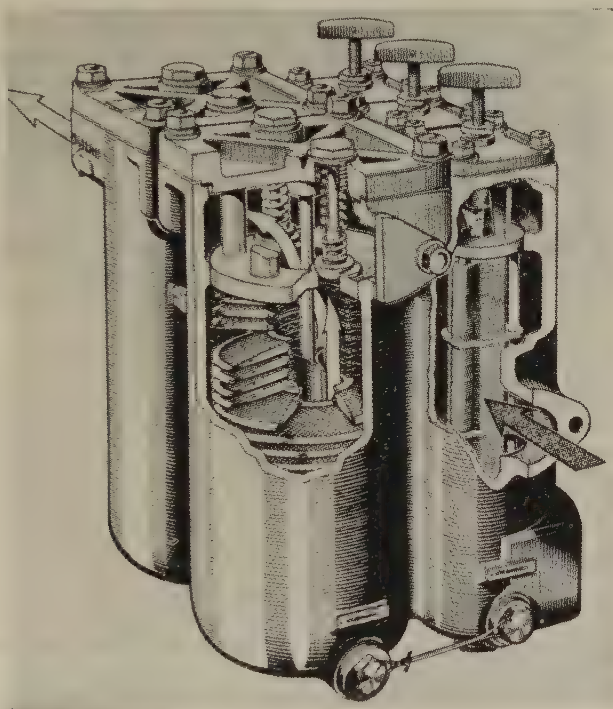


Fig. 28. — Mann lubricating oil filter.

This idea has been extended in recent years as a result of laboratory tests [3], which demonstrated that this total slip linked with the known high values of the coefficient of adhesion did not really occur when rolling freely except after a certain wheel diameter or rolling slip had been exceeded, whereas a smaller rolling slip is more or less absorbed by the elastic deformation of the materials concerned of the rail and the wheel at the point of contact and that, according to the value of the friction slip, « pseudo-coefficients of adhesion » of much lower importance must be used. Subsequently, the Deutsche Bundesbahn measured the couples set up in the cardan shafts of a transmission of a bogie with coupled axles when carrying out tests with the haulage of trailer vehicles. The measurements confirmed fully that this is the case on a railway vehicle [4].

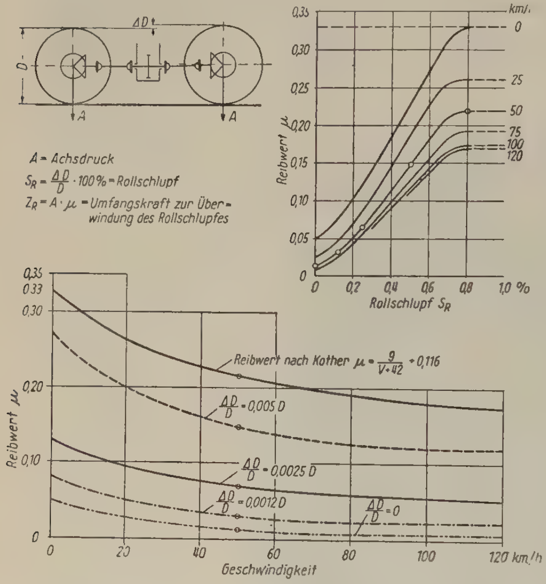


Fig. 29. — Rolling friction and slip between wheel and rail. *At the top, left :* diagram showing two pairs of driving wheels coupled together of different diameters D and D + Δ D; *below :* variation of the value of μ for different values of the difference of diameter $\frac{\Delta D}{D}$ and of the running speed V; *at the top, right :* variation of the value of μ as a function of the rolling slip for different constant running speeds.

A = load per axle.

$S_R = \frac{\Delta D}{D}$ 100 % : rolling slip.

$Z_R = A.\mu$ = tangential force needed to overcome the rolling slip.

N. B. — Reibwert = coefficient of adhesion. — Rollschlupf = rolling slip. — Geschwindigkeit = speed.

Figure 29 (at bottom) indicates for various differences in diameter Δ D/D = 0; 1.2; 2.5 and 5 ‰ « pseudo coefficients of adhesion » $\mu = \frac{Z}{A}$ in terms of the running speed V. In figure 29, upper right, are shown separately the influence of the difference of diameter at various constant speeds. It will be seen that for differences of diameter of 1 to 2 ‰, the coefficients of adhesion are only a fraction of the values expected in the case of total slipping. Furthermore, it is necessary to take into account the fact that even with exactly equal diameters, the coefficients of adhesion are found to be of the same order of magnitude. Total slipping only occurs when the difference in diameter exceeds about 7 ‰ and naturally the condition of the rail plays its part according as it is dry, humid, or sanded.

Now, as everyone knows [4] when

(3) JAHN : The relations between the wheel and the rail, *Z. VDI*, 1918, p. 121.

SACHS : Tests on the friction of solid bodies, *Z. AMM*, 1924, p. 1.

FROMM : Calculation of slip during the rolling of deformable discs, *Z. AMM*, 1927, p. 27.

LORENZ : Rail and wheel, *Z. VDI*, 1928, p. 173.

(4) GÖSSL : Installation, stresses, and trial results, with transmissions with cardan shafts on locomotives and railcars, *ETR*, No. 10, October 1954, pp. 428-430.

repairing steam locomotives after 100 000 to 150 000 km in service, the differences in diameter of the wheels on the average does not exceed 1 $^0/_{00}$ which means that in reality the wheels wear equally and that the power used through this cause can only be a fraction of that expected.

When these results were known, there was no further opposition to the simple multiple drive as such that described in Chapter III, § 2, the more so as for other reasons such as for example the possible sub-division of the power to make it easier to transmit the power need not be considered in the case of the purely hydraulic VOITH transmission of which there was already in use a proved design able to deal with the considerable power developed in multiple axle drives. As in any event the power was to be divided between two engines, it was logical to fit each bogie with its own hydraulic drive incorporating its reverse gear and to drive the three axles coupled through cardan shafts and axle gears. In this way, the two bogies are independent and in the event of possible replacements it is only necessary to see the wheels of one bogie are turned with the same diameter. In addition, it is also possible to run with only one engine set in use (fig. 30).

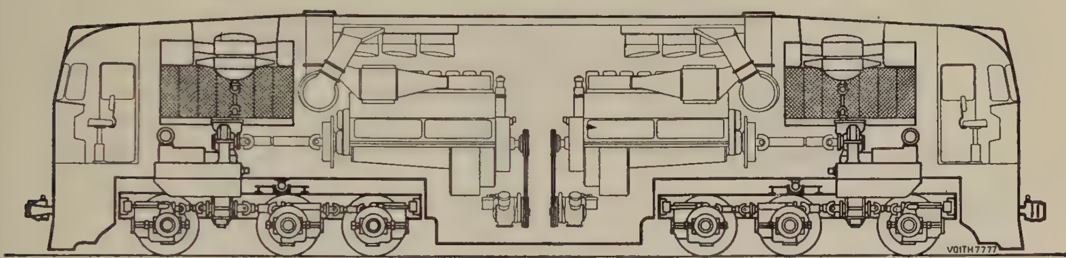


Fig. 30. — Layout of the transmissions. Two VOITH hydraulic transmissions. Six VOITH axle drives and two sets of VOITH coolers.

2. Hydraulic transmission of the power.

In order to get the specified tractive performances, all six pairs of wheels must be driven in order to have the necessary adhesive weight. Moreover, to make full use of this adhesive weight, that is to say, in order to attain when starting the tractive efforts corresponding to the limit of adhesion, a type L 36 VOITH transmission was selected with three torque converters. In this way, it was possible to make the speed at full power to correspond with the working of the third torque converter so that even under part loading it was possible to operate at the maximum speed with a reasonable efficiency.

3. Construction and method of operation of a converter.

In a hydro-dynamic torque converter, the heart of the hydraulic drive, there are a centrifugal pump and a turbine arranged in a closed circuit in accordance with « Föttingers » invention. There are as figure 31 shows no losses through a spiral casing, bends, and passages. The couple passed from the motor to the pump is transformed in the converter in such a way that for a more or less uniform power taken by the pump-wheel, there is, available at the turbine shaft, a high torque at a low speed of rotation or at a high speed of rotation a correspondingly smaller torque. For this conversion to be possible, a fixed guiding wheel must be installed between the pump wheel and the turbine able to absorb the difference in torque.

In order to explain the method of functioning, the right hand side of figure 31 shows the phenomena of flow in the vanes shown in section of the turbine at different moments: when starting, at normal speed, and at maximum speed.

Let us suppose the pump-wheel to be driven at constant speed, it then projects the fluid on to the turbine in the direction of the arrows. So long as the turbine remains stationary, the liquid is violently deflected in the passages formed by the blades which are markedly curved to the rear. This sets up a couple in the turbine many times greater than that absorbed by the pump-wheel. Under this couple the turbine begins to revolve. The more the speed of rotation increases, the deflec-

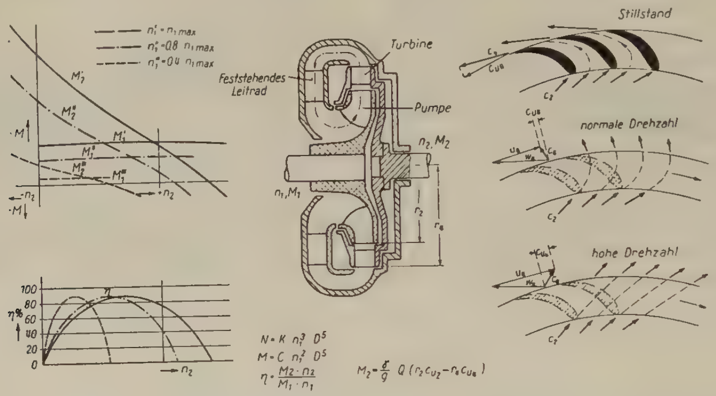


Fig. 31. — Layout and method of operation of a Föttinger torque converter. *Top, centre :* layout; *top, right :* flow through the turbine under different working conditions; *top, left :* variation in the torques and efficiencies for different speeds of the pump and of the turbine.

Symboles :

M_1 = torque at the pump;

M_2 = torque at the turbine;

n_1 = speed of the pump;

n_2 = speed of the turbine;

N = power absorbed;

η = efficiency;

Q = quantity of liquid in motion;

γ = specific weight;

c_2 = absolute speed at wheel-pump exit;

c_{u2} = tangential component of c_2;

c_4 = absolute speed at wheel-turbine exit;

c_{u4} = tangential component of c_4;

u_2 = tangential speed at wheel-pump exit;

u_4 = tangential speed at wheel-turbine exit;

w_4 = relative speed at the wheel-turbine exit (in relation to the passage formed by the turbine blades);

D = diameter of the converter casing.

K, C = constants corresponding to a given state of working $n_2 - n_1$, constant (C) = 716.2 K. K represents the horse power absorbed for D = 1 m and a speed of rotation of n = 1 t/mn.

N. B. — Feststehendes Leitrad = fixed directing wheel. — Stillstand = stopped. — Normale drehzahl = normal speed of rotation. — Hohe drehzahl = high speed of rotation.

tion and therefore the slowing down of the liquid mass becomes so much the less until as the figure shows at the bottom right at the speed of racing there is no longer any deflection of the stream of liquid. The curves on the left of the figure for constant speeds of rotation of the pump-wheel show that as the speed of rotation of the turbine n_2 increases, the couple M_2 diminishes progressively until it becomes zero at the maximum speed. This change of couple has the same characteristics as for example that of the direct current traction motor. The speed of rotation adjusts itself automatically to the resistance.

The mass of liquid leaving the turbine passes with a widely varying speed to the fixed directing wheel but, in the VOITH convertor shown, passes on to the pump-wheel without changing direction. Due to this, the couple M_1 absorbed by the pump in the different operating cases is practically constant, the speed of the pump remaining unchanged. Should the speed of rotation of the pump vary, and for the same working case, that is to say, for the same ratio between the speed of rotation of the pump and that of the turbine :

a) the power transmitted varies at the cube of the speed of rotation of the pump;

b) the couple transmitted as the square;

c) the corresponding speed of rotation directly in proportion thereto.

In the figure the curves of the couples which may be transmitted for three pump speeds are shown. These are, full speed, two thirds, and one third, thereof. It is easy to see that by altering the speed of the pump or the speed of rotation of the driving motor, the couple the converter can transmit can be varied between wide limits. We may add too that when the speed of rotation of the pump changes, the curve of efficiency follows that of the couple as for example in case of reduction of the speed of rotation of the pump or of power absorbed at the low speeds of the turbine.

The efficiency of the torque converter depends — as in all hydro-dynamic machines — upon the dimensions, the surface finish of the blades, and the viscosity of the fluid used. In addition, the ratio of the turbine speed to that of the pump for which the torque converter is designed, that is to say, for which its efficiency should be the maximum also affects the result. Depending upon the case, efficiencies of 90 % and over are realised.

4. Construction of the hydraulic transmission.

Figure 32 represents a VOITH L 36 r transmission with three torque converters. At the input end is a speed raising gear driving the common primary shaft on which are fitted the pump-wheels (cross hatched) for the three torque transformers I, II and III.

This speed raising gear is used to reduce the dimensions of the converters to small enough sizes by raising the speed of rotation. It also, thanks to the careful selection of the best ratio of multiplication, makes it possible to adopt any given transmission to suit very different powers and speeds of Diesel engines.

The turbine-wheels of the converters are permanently coupled to the common shaft of the output gear. Converter No. I is designed for starting and low running speeds and converter No. II for medium speeds. The blades of these two converters are differently arranged so that the most favourable range of working of converter II corresponds to a speed of rotation of the turbine or in other words to a train speed higher than that of converter No. I. Thanks to this arrangement the turbines of these two converters can drive the same gear. Converter No. III has the same blading as converter No. II but its turbine drives through a special gear with a smaller reduction than

that of converters Nos. I and II so that converter III only works at its best efficiency at the highest train speeds.

By bringing into action the different converters, the working ranges follows one another and are complementary and so provide a satisfactory and continuous variation in the tractive effort and in the efficiency, as shown in figure 32 (right hand).

5. Operation of the hydraulic transmission.

As already described the three torque convertors are permanently connected from the primary (pump) side to the Diesel engine and on the secondary (turbine) side with the driving wheels of the vehicle. They are brought into or taken out of service by filling or emptying the transmission, a simple method applicable without difficulty and with the greatest reliability of operation in service even up to the highest powers.

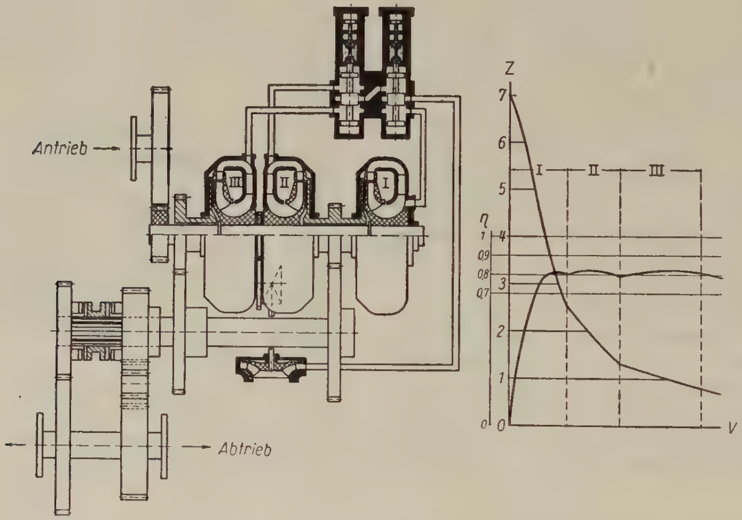


Fig. 32. — VOITH hydraulic transmission with three torque converters. *On the left :* diagrammatic arrangement; *on the right :* tractive effort-speed diagram.

N. B. — Antrieb = input. — Abtrieb = output.

In order that the vehicle may operate in both directions, the shaft of the speed reducing gear on the output side can be connected to the output shaft properly speaking of the transmission through a pair of gear wheels, either by direct drive or by a dog clutch which can be operated when stationary by means of a compressed air servo-motor. The high speed gear is directly connected by a gear wheel driving a vertical shaft which drives through cardan shafts the fan of the cooler group located above the hydraulic transmission (fig. 30).

The different circuits are filled by a centrifugal pump. The transmission is purely hydraulic and has no mechanical clutch such as dogs, frictional clutches, free wheels, etc., with the single exception of the dogteeth of the reverse gear which can be operated only when stationary and subject to suitable safety precautions.

Filling and emptying the different cir-

cuits have to correspond to certain conditions from an operating angle; they are enumerated below with the steps to be taken to carry them out.

a) *Filling speed.*

The filling pump is designed with ample capacity; from its discharge-pressure characteristic, it discharges a large quantity of oil when filling begins because there is no back pressure. The back pressure only increases as the circuit fills and the discharge from the pump falls to the amount needed to maintain the constant circulation through the circuit and make good any leakage. Moreover, in order to fill the circuits quickly an effective device has been perfected to remove the air and to help the circuits empty, an arrangement to admit air. Figure 33 shows that these measures have been totally successful. In about one second from starting to fill, convertor I absorbs the full power. This result would appear to meet all practical needs.

b) *Prevention of the formation of vapour (cavitation).*

Vapour if formed would be a nuisance and is prevented by the pressure of the filling pump superposing itself on the phenomemon of circulation of the fluid and by the latter being continuously in motion in the circuit. The objections occasionally raised in this connection against hydraulic transmission are without substance in connection with the VOITH principle of changing speed by « filling and emptying » (5).

c) *Avoidance of interruption in the tractive effort when changing gear.*

The emptying of one of the circuits and the filling of the next one take place simultaneously. The success of this method is shown in the diagram of speed changes for a transmission of three torque converters shown in figure 33. It will

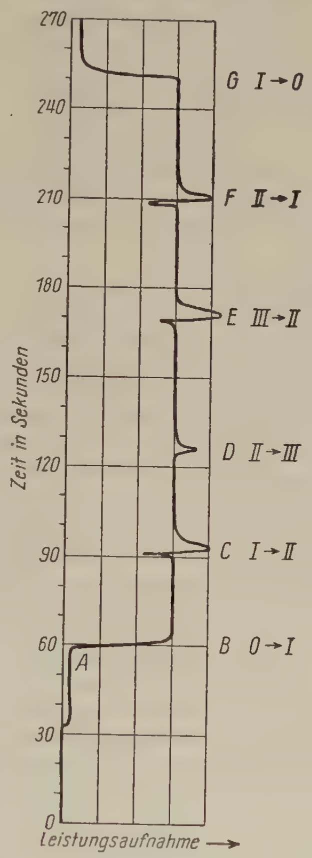


Fig. 33. — Starting diagram taken from the tape of a recording watt meter with a VOITH hydraulic transmission with three torque converters showing the power absorbed when bringing into action the converters. Converters I, II, and III; 0 = running light.

N. B. — Zeit in Sekunden = time in seconds. Leistungsaufnahme = power absorbed.

be seen that in passing from converter I to converter II, at C, and from convertor II to convertor III, at D, as when in the reverse direction, there only occurs

(5) LANG : Considerations on the transmissions of power for railway motor vehicles, *MTZ*, No. 2, October 1946, p. 17.

BRETSCHNEIDER : Hydro-static transmission in the construction of motor vehicles, *ATZ*, No. 3, March 1953, p. 80.

a small change in the power absorbed and therefore in the tractive efforts. It must be remembered too that when considered in conjunction with the Diesel engine, these variations are still less appreciable because when the load increases the speed of the Diesel falls momentarily and increases when the load falls off. When undergoing tests at the Esslingen testing plant, where too the weight of the train was not brought into account, the changes of gear were not noticed at the dynamometer measuring the tractive effort and only caused a slight fluctuation at the tachometer.

d) *Driving the empty circuits.*

As there is never more than one of the three circuits full, the other two have to revolve empty. Nonetheless, this does not mean any appreciable loss of power as is shown by the simple theoretical consideration that the work done in circulating the air is determined by the ratio of the specific weight of the fluid in the transmission to that of air. That of air being 1.29 kg/m³, that of oil 860 kg/m³, the power absorbed in the empty circuit will be 0.15 % of that when full of oil. As two torque converters are always revolving empty — when superficially considered — the power absorbed by revolving in air must be double that in the primary circuit. It must not be forgotten, however, that this energy in the air is more or less useful energy for the secondary converter so long as the turbines are not running faster than their racing speed (fig. 31). This is the case so long as the locomotive is operated on one of the two first converters, when the average loss will amount to only part of the power absorbed in revolving in the air. It is only when considering the field of operation of the third convertor when first the first turbine, then near the maximum speed, also the second turbine, are revolved at a speed higher than their racing speed that a progressively increasing resistance against revolving in air shows itself but which taking the air resistance in the primary circuit into account, at the maximum speed only reaches a value of 1 % of the power absorbed by the oil.

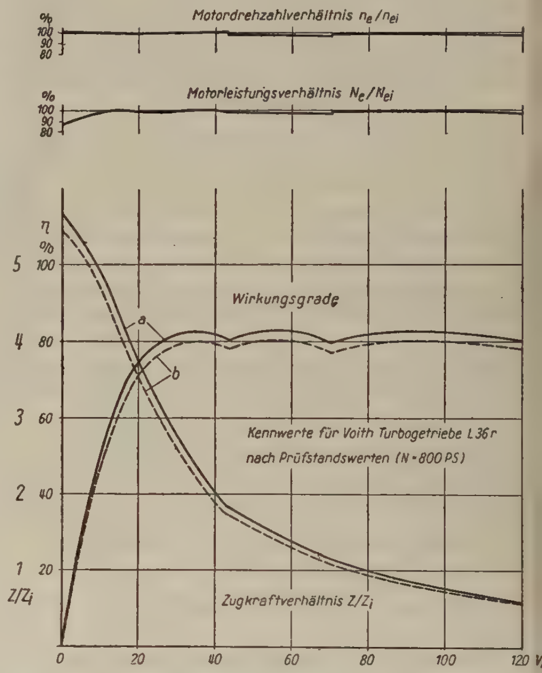


Fig. 34. — Tractive efforts and efficiencies of a VOITH hydraulic transmission with three torque converters: *a*) tractive effort and efficiency related to the output side of the hydraulic transmission (at the test plant); *b*) tractive effort and efficiency related to the driving axles : Z = tractive effort; Z_i = N.270; V_x = tractive effort in the case of a transmission with no loss when V/V_x = 100 %; V/V_x = relative running speed.

N. B. — Motordrehzahlverhältnis = ratio of the speeds of rotation of the engine. — Motorleistungsverhältnis = ratio of the powers of the engine. — Wirkungsgrade = efficiencies. — Zugkraftverhältnis = ratio of tractive efforts. — Kennwerte für VOITH turbogetriebe = characteristics of the VOITH L 36 *r* hydraulic transmission from values obtained at the test plant (N = 800 HP).

The feed to the different converters is controlled by piston valves which are moved to the correct position by a valve gear not shown in figure 32. This

arrangement is used to ensure the converter brought into gear operates under the most favourable conditions as regards running speed and power at the moment. As already mentioned in connection with figure 31, the efficiency curve of the converters moves with the reduction of power due to the drop in speed of the engine towards the area of low speeds. As the change from one to another con-

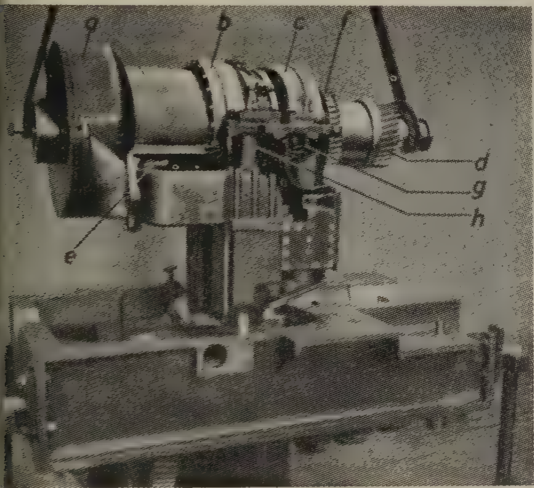


Fig. 35. — Voith hydraulic transmission with three torque converters; transmission open, rotor lifted up.

a = torque converter I;
b = torque converter II;
c = torque converter III;
d = input pinion for the pump shaft common to converters I, II and III;
e = output pinion for converter I and II;
f = output pinion for converter III;
g and h = valves of the piston distributor.

verter ought always to occur near a point of intersection of the efficiency curves corresponding to the different converters (fig. 31, to the right), the change of coupling at reduced power should be made at a lower running speed. The preselector should be affected therefore both by the power of the engine and by the running speed. In practice, this is ensured by using a small control pump driven by the engine and which gives a pressure when discharging through a measuring venturi tube varying as the square of the speed of rotation. A second pump is used in the same way and is driven off the locomotive driving axles: the pressures produced vary with the running speed. The pressures from these two control pumps act on the two faces of one piston in the preselector gear unit so that its position, and consequently, the incoming oil under pressure for moving the piston valves is determined by the ratio between the two pressures which correspond to the running speed and speed of rotation of the engine. Special steps were taken to ensure the piston of the preselector changes its position brusquely. Furthermore, to prevent oscillations a certain hysteresis was introduced between the operations of changing gear. For example during increasing speed, the next higher converter is brought into action at a higher speed than when changing down to under decreasing speed. The different converters therefore are brought in to gear or cut out of gear in an entirely automatic manner.

Preselection is brought into action or cut out from the driving post but as a whole only. This operation is automatically coupled to the regulation of the speed of the engine in such manner that when the regulator handle is in the light running position, the engine runs at idling speed. At the same time, the hydraulic transmission is entirely free, in other words, all the torque converters have been emptied. In the first running position of the regulator, the motor continues to idle but the preselector comes into action and the torque converter best suited to the running conditions at the moment is filled. When the driver moves the regulator again the engine speed increases progressively up to its maximum speed and power.

The blades of the pump-wheels are fixed, which allows a simple construction avoiding wear of the converters. This

arrangement gives the characteristic curve shown in figure 34, in which the power absorbed by the transmission for a given engine speed is more or less constant whatever the running speed of the locomotive (ratio of the engine powers N_e/N_{ei}). No regulation device intended to adjust the power supplied by the Diesel engine to the power absorbed by the transmission is needed: the regulation is automatic.

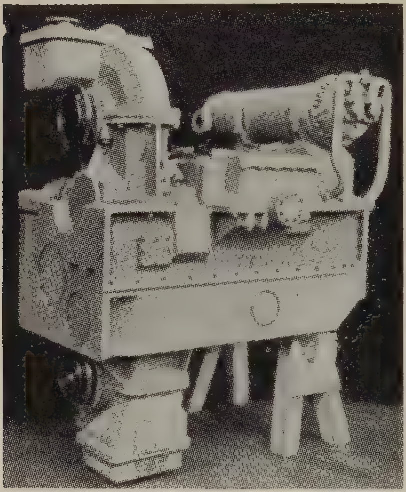


Fig. 36. — Outside view of the VOITH type L 36 *r* hydraulic transmission with three torque converters.

In the event of two sets of equipment being controlled simultaneously as in the present case, the controls of the two hydraulic transmissions are entirely independent of each other. It is neither essential nor desirable for changes in the converters to take place simultaneously. The regulation of the speeds of the engines and the impulses into the preselector in order to bring in or cut the hydraulic transmission are carried out jointly and together by means of the regulator. The question of synchronisation between the two engines does not arise.

Figure 34 shows the variation of the tractive efforts and of the efficiency of the transmission with three torque converters measured at the input flange and at the output flange (curve *a*) and also that at the driving axle after deducting the losses in the axle drives and cardan shafts.

Figure 35 shows an open transmission from which the groups of rotors, with the filling pump incorporated therewith and the control by the piston valve have been removed and the lower and upper parts of the casing taken off.

6. **Lubrication.**

All the bearing faces of the gear teeth coming into contact, as well as the bearings, are lubricated by a stream of oil. A supplementary lubrication pump driven off the output shaft of the transmission ensures the lubrication of parts which may be revolving even when the engine is stopped.

7. **Cooling.**

The losses arising in the circuits cause a corresponding heating up of the transmission fluid. To get rid of this heat part of the transmission oil is passed in accordance with the well known method through a cooler and after being cooled returned to the oil reservoir in the transmission gear casing. As cooler a heat exchanger is employed, through which the cooling water of the engine is circulated. This water is only heated a few degrees and is subsequently cooled in an air-cooled radiator. Relatively to the direct cooling of the transmission oil by an air cooled radiator, this system has the advantage that owing to the much smaller difference of temperature between the oil and the engine cooling water the fluctuations of temperature are relatively small even when the amount of heat produced varies widely. Furthermore, as the engine cooling water is automatically kept to as nearly a constant temperature

as possible, excessive cooling of the transmission oil is avoided. By using the method of cooling the transmission oil the temperature of the transmission oil has been closely regulated under all operating conditions. The lowest operating speed, that is to say the speed at which the full power of the engine can be transmitted during a long period without

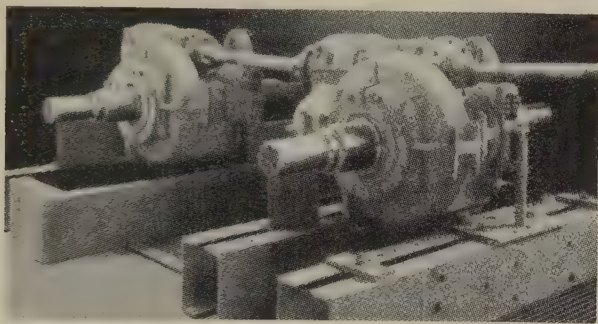


Fig. 37. — Axle drives with straight and conical gears at the test plant.

any anxiety as to excessive heating of the transmission oil, is therefore definitely lower than that allowable with electric traction with which such definite cooling conditions cannot be realised. This feature has great importance when hauling heavy loads up long steep grades.

The oil tank of about 200 l capacity required when the principle of changing speed by filling and emptying provides a supplementary heat buffer which when starting heavy trains is a precious feature of this type of transmission.

These features too help in lengthening the useful life of a fill of oil.

Figure 36 shows an outside view of the hydraulic transmission. The heat exchanger is fitted directly to the transmission casing. The transmission is carried in the bogie on three spherical bearings, one being fixed whereas the other two have a little play. By this means, any deformation of the bogie does not distort the transmission casing.

8. Axle drives.

To avoid additional losses and to reduce the distance between driving axles to the minimum, it is necessary to transmit the power from the hydraulic transmission to the couplings to the axle drives directly through cardan shafts and not, as for example, through toothed wheels from a

Fig. 38. — Transfer to the metric gauge test line.

neighbouring axle drive. Furthermore, all the axle drives should be standardised as far as possible. With this object in

view in addition to the conical gears, a straight supplementary gear in order to get over the axle to be able to couple with the next axle is needed.

In order to be able, in case of accident, to isolate the drive from the driving axle, care was taken to arrange for the conical pinion to be easily disconnected from the wheel fitted to the axle.

VI. — TRIALS AND POWER TESTS.

For testing the locomotive the Esslingen Works had on its own land a length of metre gauge track only 350 m long. This did not appear adequate for testing a 1 900 HP locomotive. The German Bundesbahn line nearby also was not suitable for testing the locomotive under all

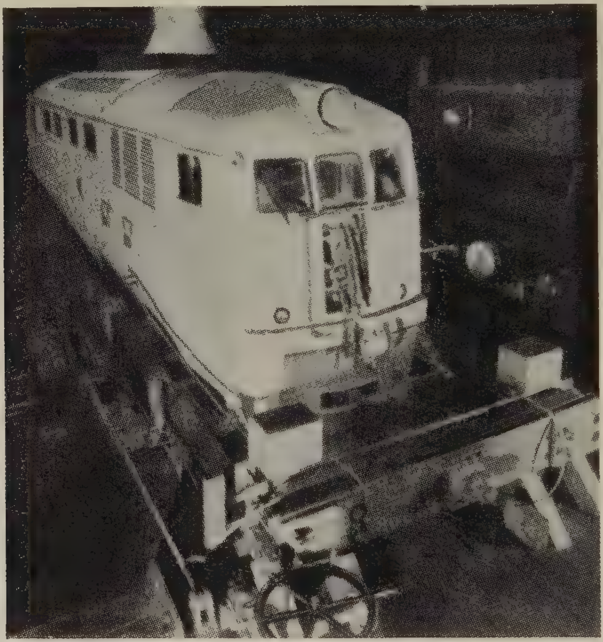


Fig. 39. — Locomotive at the test plant.

working conditions. In order to be able to see how the locomotive took the curves and also how it ran, the first locomotive was transferred to the metre gauge Nagold-Altensteig line over which it ran some 1 200 km. The standard gauge bogies used when transferring the locomotive were subsequently used to take the locomotives to the port from which they were shipped. The locomotive was put on the metre gauge line at Nagold by the Breakdown Train of the Deutsche

The lubricating oil gear pump was designed on a very ample scale and so constructed that it was not necessary to provide means for evacuating air from it : the oil being circulated as soon as the engine turned over on starting.

Figure 37 shows two axle drives coupled together at the testing plant. Amongst other features will be noticed the arrangement (turned upwards) of the torque reaction levers built into the top part of the casing.

Bundesbahn from Tübingen. Figure 38 shows one step in the transfer operation.

The trial runs in the Black Forest were almost free from any trouble. A few minor defects came to light but no doubt these would have been found during the ordinary tests in the Workshops. For the purpose of making the power tests and the other thorough trials, the testing plant developed by Lomonosoff for testing the first Diesel locomotive built for the Russian State Railways in 1924 was rebuilt at the Maschinenfabrik Esslingen.

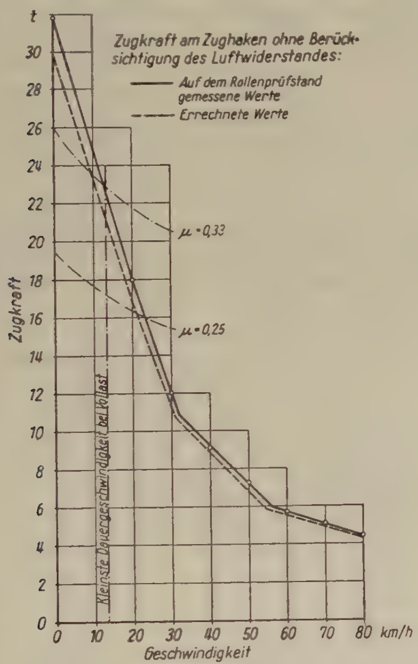


Fig. 40. — Power-speed diagram with the calculated values and those measured at the test plant.

N. B. — Zugkraft am Zughaken... = tractive effort at the drawbar hook, air resistance not taken into account; ——— values measured at the test plant; — — — calculated values. — Kleinste Dauergeschwindigkeit bei Vollast = minimum speed at full power.

Figure 39 shows a locomotive on this test plant. The rollers are braked by the water cooled blocks to be seen on the left of the figure. This brake is operated by compressed air and can be adjusted very accurately by a screw control. The tractive effort of the locomotive is transmitted through a dynamometer coupled to the draw hook to a carefully checked pressure indicator. The speed is given by a tachometer driven by a friction wheel. One bogie at a time is placed on the rollers and very detailed tests were then made of one power plant. Then the locomotive was turned round by means of two cranes and the other power plant tested in the same manner.

This method of testing was found to be of the greatest value. The cost of the installation of the test plant was made good by the knowledge acquired by means of it. The motors and the transmission had been submitted to separate thorough tests by the builders. Nevertheless, it was found that when both power plants were running at the same time certain problems arose that the separate tests had not brought to notice. It was found necessary to carry out certain adjustments to get entirely satisfactory working.

This knowledge was only acquired because the locomotive could be run under all conditions of load and at all speeds. The oscillations were measured, the cooling system was verified and the powers developed recorded. The very thorough tests to which the first two locomotives were subjected enabled the remaining defects in them to be eliminated before building the bulk of the order. The tests of the 23 locomotives i.e. of 46 power plants provided the designing department and also the staff responsible for the locomotives on delivery with much valuable information.

Figure 40 shows the curves of the theoretical tractive efforts and those measured in average for one power plant. The values measured were in all cases higher than those calculated. By using the brakes skilfully in conjunction with sanding it was found possible to reduce the speed to zero whilst the motor was

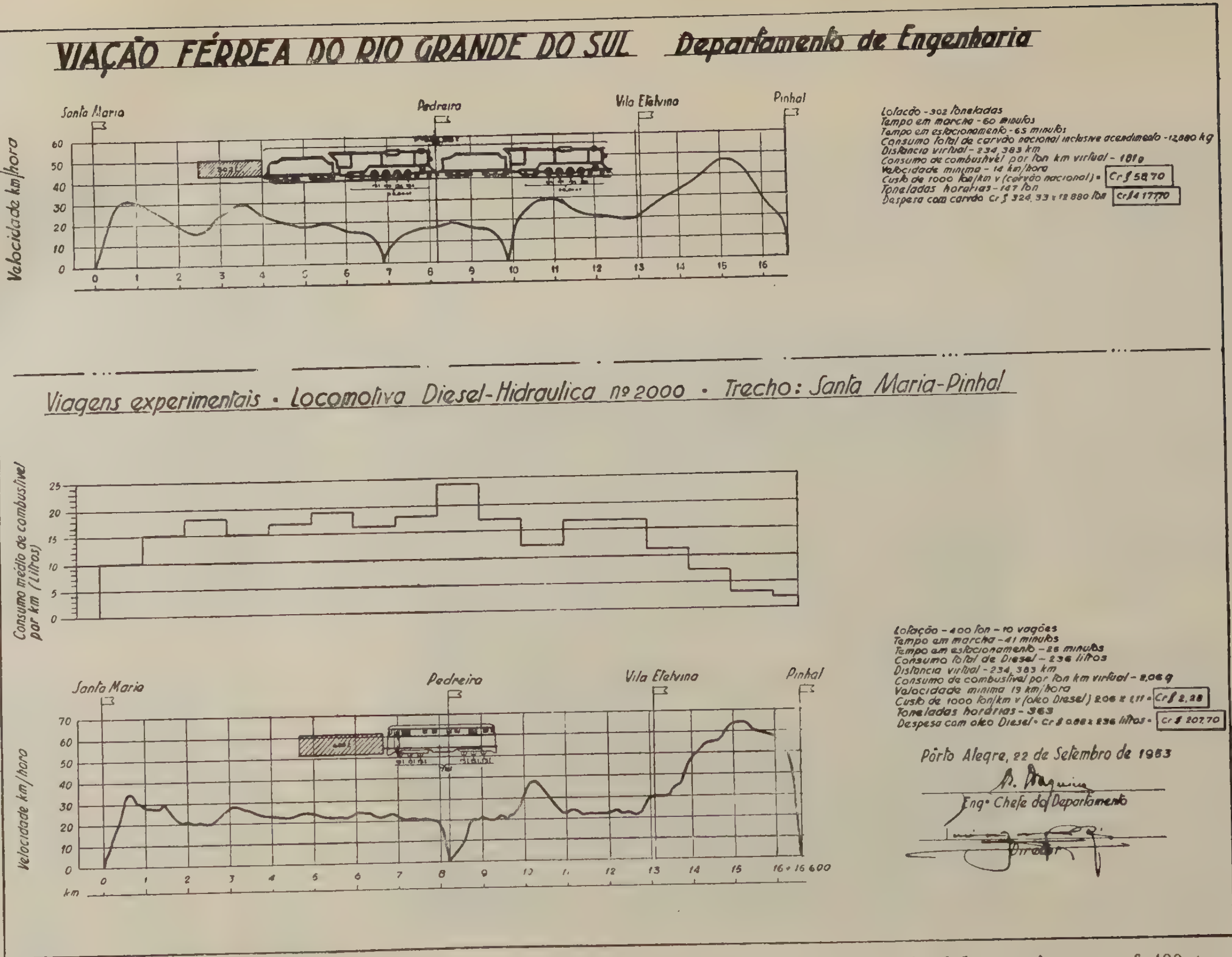


Fig. 41. — Comparison between steam locomotives hauling a train of 300 t and the Diesel locomotive one of 400 t.

N. B. — Velocidade = speed. — Consumo medio... = average fuel consumption per kilometre (liters). — Viagens experimentais... = trial runs Diesel hydraulic locomotive No. 2000 between Santa Maria-Pinhal.

Translation of the Spanish terms:

At the top:

Load : 302 t.
Running time : 60 min.
Time standing : 63 min.
Total consumption of national coal, lighting up included : [illegible]

Cost per 1 000 t/km (adjusted) (national coal) : 58.7 cruzeiros.
Tonnes per hour : 147.
Expenditure with coal : cr. 324.37 × 12.880 = cr. 4 177.70.

Below :

Load : 400 t — 10 wagons.
Running time : 41 min.
Standing time : 25 min.
Total gas oil used : 236 l.
Adjusted distance : 234.383 km.
Fuel consumption per t/km adjusted : 2.06 g
Minimum speed : 19 km (11.80 miles)/h.
[illegible] Diesel oil : cr. 2.06 × 1.11

developing its full power. The factor of adhesion calculated from the tractive effort and the adhesive weight was close to 0.40.

The changes of speed in the VOITH transmission type L 36 *r* only caused a momentary change in the tachometer reading. There was no marked change on the tractive effort.

VII. — SERVICE OPERATED IN BRAZIL.

Up to the date of this report, 20 locomotives have been brought into service and have run in all over 1 000 000 km (620 000 miles), each of the first two having covered 100 000 km (62 000 miles). The condition of the lines in the regions operated in allow an average speed of no more than 30 km (18 miles) an hour. In order to meet the guaranteed performance the V. F. R. G. S. made comparative tests with the 2′D2′-*h*2 with 3′3′T 40 tender steam locomotive, built by the American Locomotive Company, which had covered the service till then and the C′C′ 1 900 HP Diesel locomotive. The results of the line tests are shown in figure 41. It shows that two steam locomotives coupled together make the run from Santa Maria to Pinhal, 16.6 km (10 miles) long, with a trailing load of 300 t in 60 min, which corresponds to an average speed of 16.6 km/h with a minimum of 14 km (8.70 miles)/h. The 1 900 HP C′C′ only took 41 min with a trailing load of 400 t, with an average speed of 24.3 km (15 miles)/h and a minimum speed of 19 km (11.80 miles)/h. The cost of fuel amounted for the two steam locomotives to 4 177.7 cruzeiros and for the Diesel locomotive to 207.7 cruzeiros.

A comparison between the fuel costs is not sufficient to form a complete picture of the financial results obtained with steam and Diesel traction respectively. Nevertheless, the fact that two steam locomotives were required to work a train a quarter lighter shows that as regards the two factors cost of operating materials and load hauled the overall financial efficiency is in favour of the Diesel.

The comparative trials carried out demonstrated clearly the advantages to be obtained from the light weight construction of modern Diesel locomotives. Whereas the Diesel locomotive weighing 80 t in working order uses its whole weight for adhesion and does not need any carrying axles, the steam locomotive has the unfavourable ratio of its adhesive weight of 52 t to its total weight in working order of 175 t tender included. With the two steam locomotives, the weight on the carrying pairs of wheels, tender included, amounts to 246 t or about 80 % of the load hauled.

[625 .143 .3 & 625 .143 .4]

Stresses to which track equipment is subjected. Undulatory wear of rails,

by P. DUBUS,

Ingénieur à la Direction de la Voie de la Société Nationale des Chemins de fer belges.

STRESSES TO WHICH TRACK EQUIPMENT IS SUBJECTED.

In the German review *Eisenbahntechnische Rundschau E.T.R.*, No. 9, September 1954, we were advised of the results of many measurements of stresses carried out by means of strain gauges on the track on steel fishplates and the main components of rail fastenings, in particular on concrete sleepers.

These measurements were made by the Bundesbahn-Zentralamt Minden (Westphalia) under the direction of Messrs Christian BETZHOLD, Dipl.-Phys. and Helmut RUBIN, Dipl.-Eng.

These resulted in very interesting conclusions, which are summed up hereafter. During this report, mention will be made of the harmful effects of undulatory wear of the rails, which obliges certain railways to run special trains equipped to grind the rail heads.

The unfortunate consequences of this phenomenon as regards the life of the track equipment and even of the rolling stock have led us to consult technical railway publications for studies into this subject to date and we have endeavoured to classify the elements of a synthesis on this subject.

In the past, during the evolution of track equipment, the rail and the sleeper have been studied by many eminent research workers, whose studies have tended to determine theoretically the stresses due to these two important components of the permanent way.

These studies all refer to the compression of the ballast, as defined by WINKLER's formula $p = Cy$ in which C is the coefficient of the ballast. pressure in kg/cm^2 producing a driving in of 1 cm (3/8″).

It is practically impossible to take into account the harmful effects of impact, especially in line with the joints, which takes away much of the practical value of the remarkable mathematical developments of the above mentioned studies.

The fishplates and fastenings of the rails to the sleepers have not been the subject of so many theoretical investigations; the profiles and methods have evolved as a result of observing their behaviour on the track; they are above all the fruit of experience.

The method of taking measurements by strain gauges have enabled the « Bundesbahn Zentralamt Minden » (Westphalia) to sound the track in line with the joints in the case both of supported joints and overhanging joints.

Formerly, in order to determine the forces coming into play, the two mechanical relations were used, i.e. :

force = mass × acceleration;

stress = elongation × modulus of elasticity.

At the present time, as is known, strain gauges make it possible to determine the elastic stresses as a function of the variation in electrical resistances, which, in the measuring equipment, practically suppress any effect of the mass from falsifying the readings.

We will not go into further details about this equipment which today is widely used by laboratories to determine the stresses in metal bridges and other structures.

They are based upon the variations in the electrical resistance of wires made either of constantan or a nickel-iron alloy. The elongation of the wire gives rise to an increase in the electrical resistance, a shortening of the wire, to a reduction in this resistance.

The admissible field for the frequencies of stresses is very large. The frequency may vary between 0 and 20 KHz.

RESULTS.

1. Fishplates.

Stresses in the fishplates are naturally a function of the axle load, the distance between them and the speed of the trains.

In brief, deformation of the rails, in the two cases under consideration — supported and overhanging joints — takes the general form shown in the sketch (fig. 1). The inversion of the deformation of the fishplates will be noted.

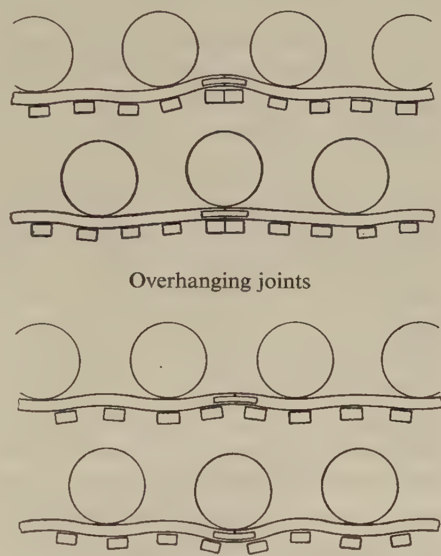
The distribution of the bending stresses in the fishplates follows the laws shown in the attached sketch (fig. 2).

2. Fastenings.

The object of these is to transmit to the sleepers the stresses acting on the rails because of the rolling loads.

These reactions can take any direction in space.

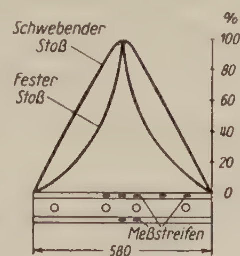
Supported joints



Overhanging joints

From E.T.R., September 1954.

Fig. 1. — Deformation at various positions of the load.



From E.T.R., September 1954.

Fig. 2

N.B. — Schwebender Stoss = overhanging joint. — Fester Stoss = supported joint. — Messtreifen = strain gauges.

The ideal in this transmission of stresses would be for the fastening holding the rail to the sleeper not to be subjected to any dynamic stresses, especially in the vertical direction, which stresses are often of high frequency, which jeopardizes its resistance to repeated stresses. The problem has been studied in both cases : rigid fastenings, and elastic fastenings. Whether the rail is laid directly on the wooden sleeper for example, or with a bearing plate between it and the sleeper, the phenomenon remains the same.

The foot of the rail or the bearing plate presses into the sleeper as the wheel passes over it. The spikes or coachscrews do not follow this movement exactly, as the rail, which is subject to vibrations, in righting itself, imparts shocks to the fastenings.

These stresses are the more harmful as the undulatory wear of the rails is more pronounced. German engineers have studied the case of elastic fastenings (elastic washers, elastic coachscrews or the insertion of a rubber bearing plate, which is also done in France).

The latter case has been carefully gone into in the case of prestressed concrete sleepers in use in Germany under two hypotheses : rails with or without undulatory wear.

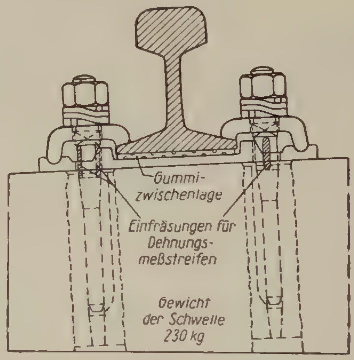
The variations in tension in the fastening spikes and in the coachscrews have been investigated.

In the case of these factors, it has been found that there is an increase in the tension when the rail has any considerable undulatory wear, which causes repetition of the stresses at a more or less high frequency.

The study which appeared in the *Eisenbahntechnische Rundschau* is full of diagrams illustrating the results obtained, and we cannot do better than give the text of their conclusions :

In the case of the fishplates :

1) The importance of the bending stresses in the case of flat steel fishplates of the German type Fl 41/49 varies with the type of joint (supported joint or overhanging joint), the position of the load, the load per wheel and the wheelbase of the vehicles.



From E.T.R., Septembre 1954.

Fig. 3. — Device for fixing the rail to the concrete sleeper.

N.B. — Gummizwischenlage = rubber insert. — Einfräsungen für Dehnungsmesstreifen = machining to enable strain gauges to be fitted. — Gewicht der Schwelle 230 kg = weight of the sleeper 230 kg.

The total amplitude of the oscillations of the bending stresses is of the order of 12 to 13 kg/mm² (7.619 t to 8.254 t per sq. in.) for an average load of 5 metric tons per wheel and a wheelbase of 5 m (16′ 5″) (Table I).

2) The bending stresses are considerably reduced when the fishplates are loosened.

3) The diagram in function of time of the bending stresses takes the form of a pointed cycloide.

TABLE I.

Amplitude A of the variation of stresse S in kg/mm² in the middle of the fishplate type FI 41/49 under rolling loads.

	Supported joint continuous bearing plate			Supported joint separated bearing plate			Overhanging joint		
	σ_I	σ_{II}	A	σ_I	σ_{II}	A	σ_I	σ_{II}	A
Locomotive R = 10 t; *a* = 1.8 m	— 1	—18	17	—17	—35	18	+16	—2	18
Passenger car-express train R = 10 t; *a* = 3 m	+ 7	—17	24	— 6	—29	23	+25	—3.5	28.5
Goods wagon R = 10 t; *a* = 5 m	+18	— 8	26	+ 6	—18	24	+25	—1.5	26.5
Locomotive R = 9 t; *a* = 1.8 m	— 1	—16	15	—15.5	—31.5	16	+14.5	—2	16.5
Passenger car-express train R = 5 t; *a* = 3 m	+ 3.5	— 8.5	12	— 3	—14.5	11.5	+12.5	—2	14.5
Goods wagon R = 5 t; *a* = 5 m	+ 9	— 4	13	+ 3	— 9	12	+12.5	—1	13.5

Legend : + Concavity, up direction. — Concavity, down direction.

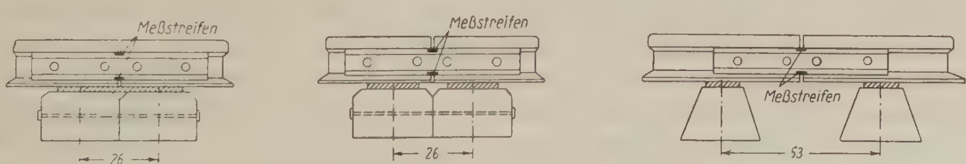
For the rail fastenings :

1) Contact between rail and sleeper is maintained, even in the case of severe vibrations, thanks to the double elastic effect obtained with the type of fastenings used.

2) The importance of the prestressing and the variations in the stresses due to the trains in the fastening bolts agrees with the values calculated theoretically and makes it possible to count upon a satisfactory life for the components in question.

3) There are no additional dynamic effects if the bed of the track is of average quality and if the rails have no undulatory wear.

4) Harmonic vibrations only occur in



From E.T.R., September 1954.

Fig. 4. — Types of fishplates studied.

the case of rails affected with pronounced undulatory wear.

5) If the rail fastenings are satisfactory and the elasticity of the parts assures control of the stresses, the influence of harmonic vibrations on the life of the fastening is very small.

UNDULATORY WEAR OF THE RAILS.

In a study on undulatory wear which appeared in the *Génie Civil*, Mr. FRÉMONT mentions two types of wear : wear caused by the sliding of the wheel, which then causes abrasion of the metal, and wear produced by deformation from shocks, which then cause local crushing. The first type of wear was reported as far back as 1906 by Mr. DUBS, Manager of the Marseilles Tramways. He expressed himself as follows at the Milan Congress :

« If the rail takes on any rapid vibration, the specific pressure and the degree of adhesion at the point of contact naturally undergo proportionate variations according to the amplitude of the vibration of the rail and are of the same frequency as the latter.

« Tangential stress at the point of contact remains constant, so that variations in the degree of adhesion may give rise to a periodic acceleration or deceleration of the angular speed of the wheel, and as the mass of the vehicle cannot follow these variations instantaneously, there must of necessity be a series of slides, the result of which will be undulatory wear of the rail. »

Undulatory wear is therefore caused by abrasion, an opinion which is also shared by the American engineers, Messrs ROBIN and DUDLEY.

Mr. FRÉMONT invented a device making it possible to take a tracing showing the distribution of the pressure on the surface of the rail when a wheel passes over it.

The pressure of the wheel on the rail does not stop, but the rail during its vibratory movement hits up against the wheel and the intensity of the shock varies according to the mass of the rail — a heavy rail gives a heavier shock than a light rail. For this reason, the fishplate joint, the mass of which may be considerable, appears to be able to start undulatory wear.

The first conclusion that comes to mind after observing these phenomena is that we should endeavour to break up the vibrations and therefore render the distribution of such vibrations as irregular as possible.

Mr. TURNER, Metallurgical engineer of the British Railways, sees as the first factor in the pronounced undulatory wear of rails a phenomenon of corrosion due to atmospheric effects. The brilliant white spots are, whatever the cause, hard martensitic zones forming protuberances. In some opinions, it is a question of a tempering action. Underneath they are separated by dark, black spots, due to the corrosion of the softer zones.

We will see what this looks like to Dr. August DRESSLER (Wien-Linz).

He is of the opinion that research should be directed towards obtaining a corrosion resisting rail steel *(Institution of Civil Engineers*, October 1954), pulverisation on the surface of rails composed of alloys with a zinc or aluminium base not being in any way a protection.

Professor FINK *(Glasers Annalen*,

December 1953) considers that undulatory wear is a physico-chemical phenomenon. From his observations and experiments it appears that under the effect of the friction due to sliding or rolling when braking or the variation in the angular speed, particularly in the case of driving wheels, a dust, sometimes even fine particles, composed of metallic oxides, is swept along the head of the rail and submitted to mechanical and chemical actions at a high temperature, which is favourable to plastic deformations. The result is the formation of the peaks and low points characterising undulatory wear.

During the course on applied mechanics which he gave towards the end of the last century at the Royal College of Science at South Kensington, Professor John PERRY when he wished to illustrate running resistance used a simple apparatus as shown below, consisting of a flat surface without initial tension made of rubber run over by an iron wheel.

Marks at intervals of an inch served as reference-marks both for the wheel and for the plane of rolling, which, says PERRY, is similar to a driving axle of a

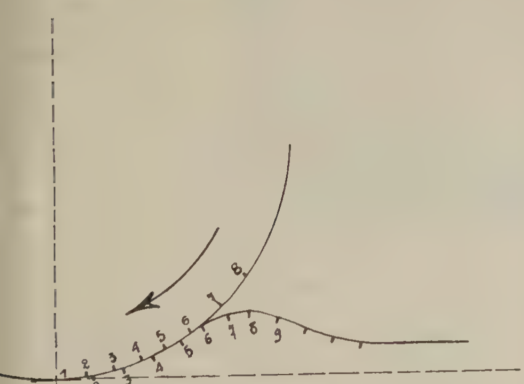


Fig. 5.

locomotive running on the rail. Experience has shown that points 1, 2, 3 and 4 draw apart from each other as points 6, 7, 8 and 9 approach each other. The energy due to molecular work is stored up both in the tyre of the wheel and in the rail, which becomes a seat for waves of compression, then of extension; the two parts in contact sometimes slide over each other when heat is generated, whence a loss of energy, less in the case of rolling alone, when the heat developed depends upon the pressure of contact, the speed of running and the radii of curvature of the surfaces in contact.

This supports the present thesis of Professor FINK.

In his study, the latter also recalls the observations made by several research workers, in particular VON SCHOTTKY and HILTENKAMP, as well as KLINGER and KOCH as regards the surface of steel submitted to mechanical actions.

It is a curious fact that the superficial zones which are mechanically stressed are enriched by nitrogenous products compared with the nitrogenous content of the internal zones.

Experiments and analyses are in hand at the University of Darmstadt under the direction of Professor Dr. HOFMANN to determine what happens in lines with the protuberances of rails showing undulatory wear on the Hambourg-Harbourg line of the Deutsche Bundesbahn.

Another more recent study is by Dr. techn. August DRESSLER (Wien-Linz).

According to him, the initial cause is the segregation often found in rolled sections. In this connection, the work of Mr. SERVAIS on the physico-chemical constitution of constructional steel includes

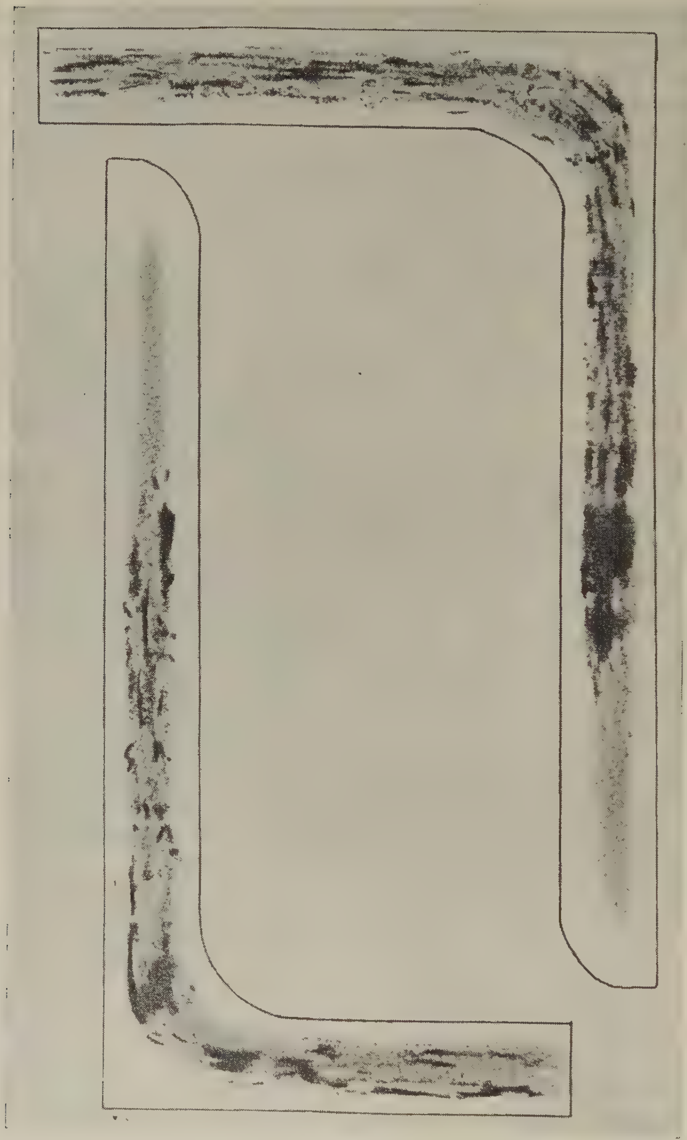


Fig. 6. — Segregation commonly found in steel sections.

some very illuminating figures (fig. 6). A segregation of the same kind can be seen in the longitudinal section of a rail given in the following sketch (fig. 7).

The author of this new theory sums up his findings as follows :

He imputes undulatory wear to a relatively simple phenomenon which

occurs when the wear of the running surface reaches the core of the head of the rail, which has a different degree of hardness to the outer layer owing to slower cooling during manufacture, and owing to the lesser resistance of the outer layer to corrosion.

The following formula can be applied to undulatory wear :

$$K_{ug} \gtreqless (V_g + K_g).$$

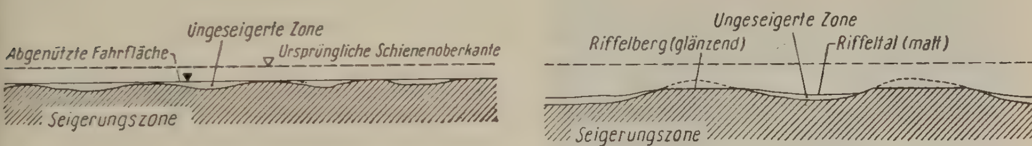
V_g being the wear and K_g the corrosion of the core, K_{ug} the corrosion of the outer layer of the head of the rail.

during the manufacture of the steel, then concerned with the rolling.

From the metallurgical point of view, an attempt should be made to reduce the zone of segregation of the section of the rail.

When the ingot is drawn, by careful gauging of the cylinders, the upper level of the zone of segregation should be lowered below the limit of wear allowed for the rail in service.

Dr. August DRESSLER's new theory is very attractive; at least he has some remedies to suggest, which it is to be hoped will be tried out.



From E.T.R., October 1954.

Fig. 7. — Longitudinal section of rail.

N.B. — Ungeseigerte Zone = Superficial layer. — Abgenützte Fahrfläche = worn running surface. — Ursprüngliche Schienenoberkante = original top of the rail. — Seigerungszone = zone of segregation. — Riffelberg (glänzend) = shining peak of an ondulation. — Riffeltal (matt) = hollow (dull).

When K_{ug} is greater than $(V_g + K_g)$, there is undulatory wear as soon as this wear reaches the core of the head of the rail.

When K_{ug} is equal to or smaller than $(V_g + K_g)$, which is the case when the rail has suffered greater mechanical wear, or when the core and the outer layer have about the same degree of resistance to corrosion, there is no undulatory wear, even when the wear reaches the core of the head of the rail.

Dr. August DRESSLER then sketches the remedies he thinks would reduce these defects.

These are first of all metallurgical, ...

The difficulty still remains intact. Grinding the head of the rails — a technique adopted by certain railways — is only a palliative, which has become necessary in order to improve the comfort for passengers and reduce the damage to the track and rolling stock.

It may also be mentioned that undulatory wear, which causes vibrations in the wheels of light railcars, may lead to a discontinuous action of the track circuits which has a harmful effect upon the proper working of the signals.

Finally, to be complete, we must mention the effects of braking and skidding.

The latter, it is known, can cause local defects in the head of the rail.

As for braking, the observations made in Switzerland by Mr. GENTON on an electric locomotive, Ae 4/7 on the St. Gothard line, show that the ordinary brake blocks cause undulatory wear of the tyres of the wheels of a very characteristic type, which does not disappear as generally agreed (fig. 8).

From Glasers Annalen, December 1953.

Fig. 8.

In conclusion, it must be recognised that we are very far from reaching agreement about the causes of undulatory wear, and consequently upon the remedies to be applied in order to avoid it.

Research work is continuing on several railways. In Belgium, the firm of FRANKI in collaboration with the S.N.C.B. and Brussels Tramways has made characteristic collections of deformations of rails. The analysis of these has been the subject of an article published in the review « Industries and Sciences ».

BIBLIOGRAPHY.

1. Spannungsmessungen an Stosslaschen und Schienenbefestigungsmitteln im Gleis, by Christian BETZHOLD, Dipl.-Phys. and Helmut RUBIN, Dipl.-Ing., *Eisenbahntechnische Rundschau*, Heft 9. September 1954.

2. A contribution to the study of rail corrugation by Ch. FRÉMONT. — *Bulletin of the International Railway Congress Association*, April 1928.

3. Applied mechanics, by John PERRY.

4. Neue Theorie über die Ursache der Riffelbildung auf den Fahrflächen der Eisenbahnschienen, by Dr. techn. August DRESSLER. — *Eisenbahntechnische Rundschau*, Heft 10, October 1954.

5. Constitution physico-chimique des aciers de construction, by J. SERVAIS.

6. Die Entstehung der Schienenriffeln. Der Stand der Riffelforschung nach rund 60 Jahren, by Professor Dr.-Ing. M. FINK. — *Glasers Annalen*, Heft 11, November 1953, and Heft 12, December 1953.

[**625** .143 .5 (45)]

Special coachscrew for railway track,

by Dr.-Ing. Giovanni ZAQUINI.

(*Ingegneria Ferroviaria*, October 1954.)

The most serious defect of the usual type of coachscrew used in rail fastenings is that it often becomes loose in the holes in the sleepers and thereby affects the stability of the anchoring it is intended to ensure.

The loosening of the coachscrews in the holes in the sleepers is due partly to the wood decaying more quickly around the bolt than in the body of the sleeper and partly to dynamic stresses exceeding the resistance of the timber.

Experience demonstrates in fact that during the service life of a sleeper the coachscrew holes have to be redrilled several times (from 2 to 4 extra holes per sleeper). This is a more or less serious matter according to the case; it means at all events a serious maintenance expenditure.

In ordinary running lines, the loosening of the coachscrews is cured by moving the sole plates along the sleepers as often as this can be done, for drilling fresh holes and plugging the old ones, and securing the fastening in the new ones.

When it is no longer possible to make fresh holes in the sleepers, these latter have to be changed even though they are otherwise still usable.

Experience shows that about 20 % of the sleepers are removed for this reason when reconditioning the track, or in other words when there is no room for new holes. The consequential financial costs are appreciable and can be evaluated as follows:

An ordinary sleeper costs about 3 000 Liras and its average service life is fourteen years.

If 20 % additional sleepers could be kept in service, their average life would increase in the same proportion, that is on an average by two years.

In view of this, the damage can be calculated per annum per sleeper laid at 2/100 of 3 000 L., i.e. at 60 L. which comes to about 78 000 L. per kilometre of track per year in service.

The slackening of the coachscrews usually causes the track to become widened, a danger for the working.

In the points and crossings, the slackening of the coachscrews is still more serious because these attachments cannot be moved along the crossing timbers. For economy, every care is taken not to move the crossing timbers or replace them.

Moving a crossing timber means drilling 16 new holes, plugging the 16 holes no longer used, as well as considerable work in removing and replacing the ballast.

The replacing of a crossing timber is very costly due to the high cost of the special type of wood and of the labour involved.

At points and crossings, especially on secondary lines, many loose fastenings are found due to a number of coachscrews turning in their holes; this is always very dangerous in railway working.

The defect in the case of the fastenings of signal or point operating mechanisms due to loose fastenings can cause serious operating trouble.

It is just for this reason that the Italian State Railways Administration tends to discontinue the practice of fastening the operating mechanism to the sleepers and instead to fasten them directly to the rails although this is much more costly.

Fig. 1. — Sleeper taken out of service recovered by using special nutted coachscrews.

The locks to hold the points in place, the operating mechanisms, the safety clips, the operating transmission of both plain and hinged points, ground signals, the disc signals, the locks holding the signals in position, etc., are often fastened to the sleeper by coachscrews.

In all these cases, the loosening of just one coachscrew can cause serious operating difficulties.

* * *

Many new designs have been tried out to overcome this defect but with little success.

A method proposed consists of a normal coachscrew slightly lengthened which when in place projects below the bottom face of the sleeper. On to this projection, on to the thread of the coachscrew is screwed a small elastic plate in steel which in all cases ensures the coachscrew is fastened to the sleeper.

In cases of gauge widening, a semi-circular packing piece (tegolino) screwed on the inner face, is placed inside the existing holes and makes it possible to restore the gauge to the correct dimension without having to redrill the sleeper.

The special coachscrew can be used with advantage in the following cases :

1) *During the upkeep of the track,* if a coachscrew is found to be turning freely in its hole according to the case the sleeper is raised by means of the usual jacks or a hollow is made under it. This done, the coachscrew is removed, the special coachscrew is then hammered into the hole, the elastic nut with its points turned up so that when the coachscrew is turned these points will be driven solidly into the bottom face of the sleeper.

If in addition to the loosening of the coachscrew, the gauge is found to be widened somewhat (often the case), before driving the new coachscrew through the hole, the semi-circular wedge piece is put into position, a rapid operation with the special tool equipment provided. This wedge piece rests in the holes of one sole plate or if need be of the two sole plates of the same sleeper.

It has been proved in practice that to

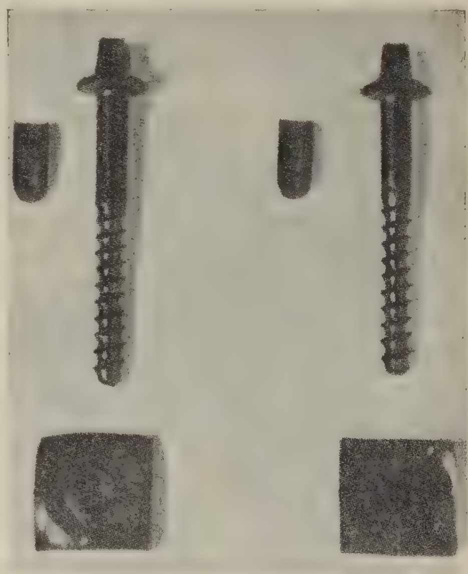


Fig. 2. — Coachscrew with nut and wedge piece.

take up increases of 1 to 2 mm (3/64" to 5/64") a wedge piece (tegolino) 3 mm (1/8") thick should be inserted in each hole of a given sleeper, taking care to set it correctly in the hole, whereas to deal with increases of 2 to 4 mm (5/64" to 5/32") wedge, pieces 6 mm (1/4") thick have to be used. Should much greater increases have to be taken up 3 mm or 6 mm wedge falling under the raised sleeper. After doing this, wedge pieces to suit the case are inserted in the holes and the long type coachscrews put in place, with the points under the sleeper as already described.

3) *When overhauling the fastenings of the operating mechanisms,* should the ordinary coachscrews not hold, the fasten-

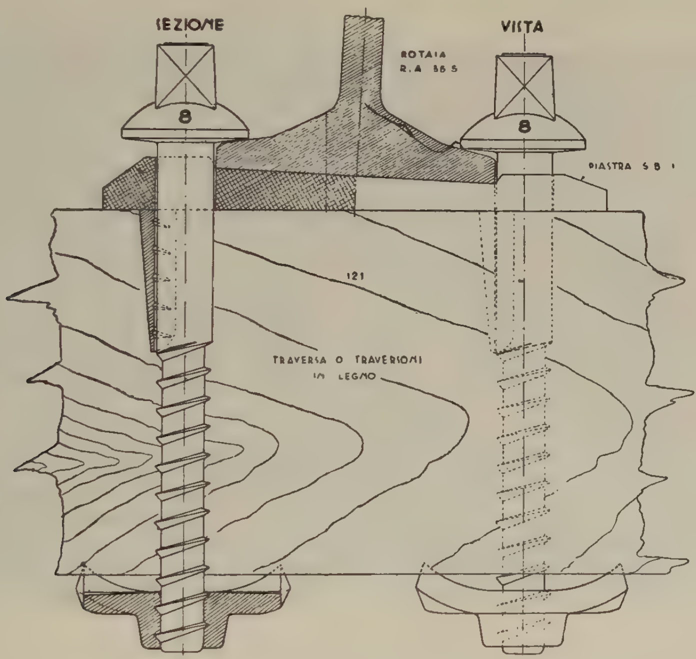


Fig. 3. — Special coachscrew in position.

N. B. — Rotaia R. A. 36 S = R. A. 36 S rail. — Piastra = bearing plate. Traversa o traversoni in legno = ordinary or crossing sleepers in wood.

pieces suitably set in position can be used in the two fastenings of the sleeper. When wedge pieces are used the coachscrew has to be screwed in.

2) *When overhauling points,* should one or several coachscrews revolve in their holes, these are withdrawn and the ballast is cleared away under the sleeper, or the sleeper is jacked up, care being taken whilst so doing to prevent the gravel from

ings are made secure by replacing them with the special coachscrews with nuts. In this way, the position of the fastening remains standard without it being necessary to change the ordinary sleeper nor the points and crossings nor to replace all the fastenings in question.

4) *When giving the track a thorough overhaul or when building new lines or new points and crossings, etc.* —The applic-

ation of the new coachscrews is simple and with trained staff quick. The through holes are drilled in the shops and the coachscrews driven in with a hammer; on the end projecting through the sleeper the elastic nut is tightened by giving the coach-screw a few turns. Indirect fastenings are secured in position in the shops but the direct fastenings at the site before the sleepers are put into place.

— weight of the elastic nut . . . 350 g
— total weight : 550 g at L. 0.20 = 110.—

b) labour :

— 5 minutes of platelayer above that taken to put in position the ordinary type of coachscrew at L. 8.0 . . . . L. 40.—

c) total cost of first laying . L. 150.—

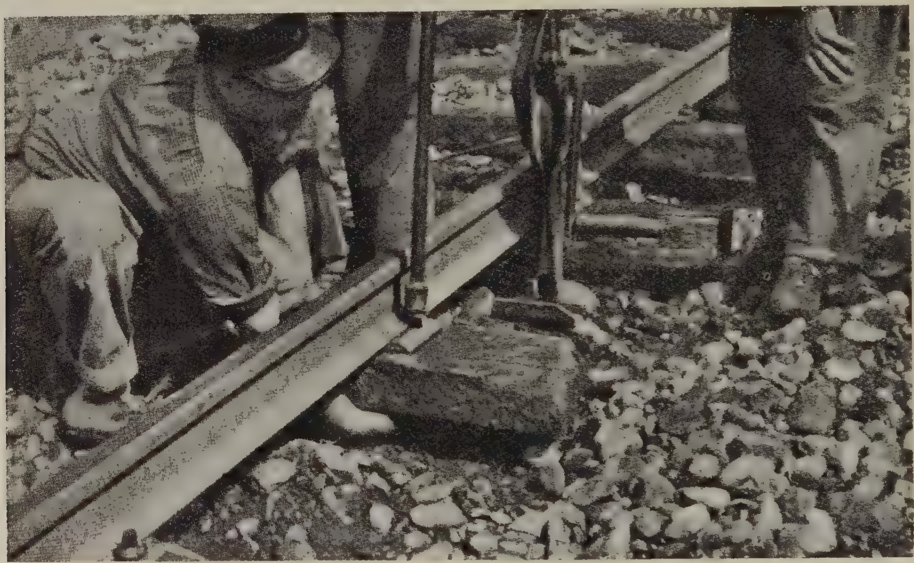


Fig. 4. — Method of applying the coachscrew.

Financial benefits resulting from the application.

On running lines during maintenance. The cost of the proposed application can be calculated as follows (case of the RAS superstructure — SB1 bearing plates — M.8 coachscrew — number of sleepers laid per kilometre 1 300) : per coachscrew of the new pattern placed in the track :

a) material :

— additional weight of the new coachscrew above that of the old . . . . 200 g

If we take into account the recovery of the material and its use again for three successive relaying, that is about 6 years, the expenditure is reduced to :

— material : 1/3 of L. 110. . . L. 36.60
— labour . . . . L. 40.—
— total cost per coachscrew . . L. 76.60

The expenditure per sleeper for an average of two coachscrews is . . . . L. 153.20

For an expenditure of L. 153.20 as an average, a sleeper which represents new and laid a value of about L. 3 000, can be

kept in service a minimum of two additional years. In other words, for an expenditure of L. 153.20 the residual value of the sleeper, about L. 600 is recovered making a saving of L. 446.8 per sleeper, by using the new type coachscrew.

During the first two years after using the new arrangement, the cost of buying two new sleepers per 100 in service, i.e.

Fig. 5. — Putting in the gauge rectifying wedge without displacing the bearing plate.

0.02 × 3 000 × 1 300 = 78 000 Liras per kilometre of track for a cost of purchase and fitting the new coachscrews of 0.02 × 150 × 2 × 1 300 = L. 7 800.

After the first two years, the financial savings become normal on new bases because sleepers with the new coachscrews begin to be replaced.

Each subsequent year, the new coachscrews have to be applied to only 1/10 of the sleepers to be replaced, i.e. to about 1/100 of the sleepers in the track. The economic benefit as has been shown of 446.8 liras per sleeper dealt with becomes : 0.01 × 446.8 × 1 300 = 5 808 Liras per annum and per kilometre of track.

— *In the case of points and crossings,* the economy is greater than on running lines, in the ratio of the cost of the different types of wooden sleeper, or in other words, between the cost of a crossing sleeper and an ordinary sleeper, the ratio of which is about 4 to 1.

— *In the case of operating mechanism fastenings,* in addition to the cheaper cost there is the greater reliability in operation.

The expenditure involved in replacing the ordinary coachscrews by the nutted coachscrews is moreover very small.

— *In the case of new lines,* the application of coachscrews with elastic nuts to new sleepers ensures the anchorage of the track during the whole life of the sleeper without any work having to be done or moving bearing plates, ferruling, etc., representing an appreciable saving in track maintenance. The use of these coachscrews is suitable especially on sections of track on curves or on track laid on sleepers of soft woods such as pine, larch eucalyptus, etc., and on sleepers having any sapwood.

Stability of earth slopes,

by J. DUBUS,

Ingénieur en chef à la Société Nationale des Chemins de fer belges.

The review « *Travaux* » has published, in the December 1954 issue, a very interesting report on the proceedings of the « European Conference on the stability of earth slopes », which took place in Sweden in September, 1954.

This report, written by Mr. A. LAZARD, Ingénieur en Chef des Ponts et Chaussées de France, Chef de la Division des Ouvrages d'art de la S.N.C.F., begins by discussing the general theory of the stability of slopes, question studied in the first section of the Congress, which had given rise to particularly interesting developments, owing to the diversity of the checking methods used.

The following is a reproduction, *in extenso*, of the part of Mr. LAZARD's report devoted to this question :

As already mentioned, the discussions on the first section of the Congress gave rise to numerous contributions which were mainly concerned with the following two points :

1) Do earth slides occur gradually or in a single movement? Are the failure lines straight or circular, or of any other shape?

2) How can « safety » be defined?

As regards the first point, it must be admitted that the majority of those present seem to be in favour of the assumption that the failure line of the sliding movements is circular, and that the slides occur in a single movement. However, Mr. FRONTARD was able to point to his research work which showed that sliding movements often occur gradually, along a failure line having the shape of a distorted cycloid. We supported this opinion by recalling that the analysis of gradual sliding movements gave practically the same results as the analysis of instantaneous movements so that the results based on the latter analysis do not prove conclusively that gradual sliding movements are impossible. Moreover, numerous contributions — including some from Swedish side — showed that it is necessary, in numerous cases, to assume failure lines which are not circular but have the shape of composite curves. The fact remains, however, that in the opinion of numerous experts present, the sliding movements do in fact take place effectively as a rotary movement around the axis of a circular cylinder. This conception (which has always appeared to us to be merely a convenient working assumption for calculating purposes) seems to be dangerous.

It is mainly on the second point that the discussions were numerous and, it must be admitted, confuse. Professor FRÖHLICH, Vienna, sought approval for his own definition of the safety factor, based on the « Additional Moment » required to bring about failure along a circular failure line, making use of all the resistance qualities of the soil. This is, therefore, a calculation of limit equilibrium. Professor FRÖHLICH's approach has not been followed, apparently mainly because his definition would not properly fit a sliding movement along a non-circular failure line, and because the results thereby obtained differ only very little from the commonly used definition formulated by Professor FELLENIUS, that « grand old man » of 78 years who did us the honour to be present during most of the discussions. Let us recall that, in this definition, the safety factor is the

ratio between the actual properties of the earth, and the properties required to prevent failure. Or, put in a different way, the reciprocal value of this safety factor is the mobilized proportion of the slide-resistance qualities of the earth. This definition which, in our opinion, is to a great extent conventional, is largely accepted by those attending the Congress as a purely logical and wholly satisfactory definition. As we shall explain later on, these differences of interpretation entail certain difficulties as regards the calculation methods.

We have tried to widen the conception of safety by introducing notions of statistics and probabilities, following the same line of argument as that followed for the safety calculation in the Bridges and Structures by the « Ecole Probabiliste Française », represented by Messrs. M. PROT and Robert LÉVI, well known to the readers of this Review. Our thesis seems to have caused some surprise, but it is certain that the idea will be followed up and will lead, in a few years time, to a perfectly logical safety calculation.

On the subject of the calculation of the safety factor, those favouring the method of regarding the sliding wedge as being cut into slices and those (including ourselves) favouring the methods of global calculation are often opposed to each other. It emerges from these discussions that the calculation method based on slices has been set out in a perfect manner, for failure lines of any shape and for any number of soils encountered, by a young Norwegian, Mr. JANBU. We hope that his work will soon be published; it takes account of all the conditions implied in the problem and leads to a succession of simple and rapidly convergent numerical calculations.

On the other hand, we have ourselves, during the very session of the Congress, prepared a complete account of the global geometrical calculation method — likewise for failure lines of any shape and for any number of soils concerned — using a method of approximation which translates into geometrical language the approximations, referred to above, of the numerical calculations. But with this method — and we believe that, in this respect, our method is at a considerable advantage —, it is possible to obtain the infinity of the combinations of the properties which the soil must have to ensure the equilibrium of a slope (whilst the numerical method, because of the definition of the safety factor, merely leads to a single combination), and thus to acquire a perfectly clear and visual notion of the stability margin of a slope consisting of soils with known properties.

After the calculation problems have now been perfectly solved, it remains to fix the safety values appropriate for each case. At the Congress, no paper has been presented dealing with this all-important point. It would nevertheless seem that the Scandinavians and British are satisfied if the stability of a natural slope (and sometimes even of an embankment) is just above 1. It seems to us that this is due to the nature of the argillaceous soil encountered in that country, and to an analytical method based on circular failure lines, which is particularly well suited to these cases. It is in this connection that, outside the proceedings of the Congress, our Swedish colleague Mr. FELLENIUS (the son of the Professor) showed us some work designed to raise the conventional safety factor from a value very close to 1 (e.g. between 0.98 and 1.01) to slightly higher values, being content with 1.10 or 1.14. Experience supports these cases. In the case of French types of clay, however, or for earth dams, it seems to us that one should not be content with such low values. We hope that progress will be made in this matter between now and the next Congress.

Other interesting questions have been debated concerning the isotropy of glacial clays and slides in sand but it has not been possible to reconcile the conceptions put forward.

Mr. BRINCH-HANSEN, Denmark, author of a valuable recently published book on retaining walls, made a contribution indicating some possible composite failure lines but with a supplementary cinematic condition which did not seem to be very clear to us.

Mr. ZAHNLE (a Frenchman working in Belgium) presented a report on calculations made to determine the stability of an earth dam, where methods almost identical to ours have recently been employed.

The stability of slopes in cuttings has, at all times, been a preoccupation of the engineers. In France, VICAT has, in 1833, published a study in the « *Annales des Ponts et Chaussées* » leading to practical formulas which he calls « Navier's formulae ». The works of Messrs. FRONTARD and CAQUOT are well known for their mathematical strictness.

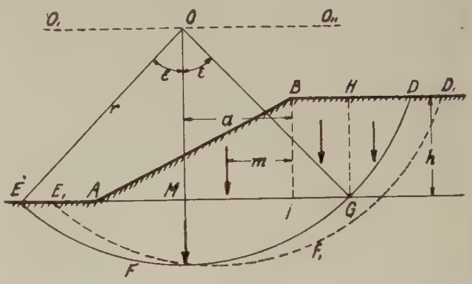
The Swedish method consisting in replacing the theoretical failure lines by arcs of a circle has, as Mr. MAYER observes in his book « Sols et Fondations », the merit of simplicity although it is perhaps less satisfactory for the mind.

In Sweden, in particular, this method has often been endorsed by experience.

Professor Wolmar FELLENIUS has set out this method in a treatise « Erdstatische Berechnungen ». At Delft, in Holland, Professors Ir. A. S. KEVERLING BUISMAN and Ir. T. K. HUIZINGA have, in their lectures on « Grondmechanika », given a detailed account, with proof. In the United States, Professors TERZAGHI and TAYLOR devote to it extensive commentaries in their lessons at the « Massachusetts Institute of Technology »; so do Professors P. Leonard CAPPER and W. FISHER CASSIE in Britain.

Some readers may be interested to know the proof given by the author of the method, Wolmar FELLENIUS (on page 13 of his treatise where he first deals with cohesive masses without static friction).

Let E.F.G.D. be a cylindrical surface tangent to the horizontal plane of the resisting formation below. The radius r and the angle at the centre, 2ε, are provisionally assumed.



In order to find the most dangerous failure line, the centre O is moved along the horizontal line O_IO_{II}.

A moments equation around the centre O yields the resistance k due to the cohesion of the argillaceous mass, ensuring the equilibrium following the circular arc EFGD :

$$k = \frac{\text{moment EFGA} + \text{moment HGD} + \text{moment BIGH} + \text{moment ABI}}{r \times \text{arc EFGD}}$$

The moment due to the circle segment EFGA is zero; that of the mass with triangular cross-section HGD is constant if the centre O is moved, say, to the right; also, the length of the circular arc EFGD remains constant during such a movement.

It is therefore sufficient to find the condition for which the moments due to BIGH and ABI attain a maximum so that, if γ represents the specific weight of the soils, one obtains :

$$\gamma(\text{ABI})(a - m) + \gamma h(r \sin \varepsilon - a)\left(a + \frac{r \sin \varepsilon - a}{2}\right) = \text{max.}$$

or

$$(\text{ABI})(a - m) + \frac{h}{2}(r^2 \sin^2\varepsilon - a^2) = \text{max.}$$

By differentiation in relation to a :

$$(\text{ABI}) - h \cdot a = 0 \qquad \text{hence : } a = \frac{(\text{ABI})}{h}$$

It will thus be seen that the centre O is located on the vertical line passing through the mid-point M of the line of the greatest pitch of the slope.

It may be mentioned that Messrs. FELLENIUS and HUISINGA also show that the angle 2ε has the value $133°34'$.

With these simple criteria determining a first centre for the circular failure line, it is possible to extend the examination to other adjacent centres and to find the relation F between the resisting moment and attacking moment on the basis of a laboratory study of the characteristics of the soil concerned; the critical cross-section is that with the minimum value of F.

Professor TAYLOR of the Massachusetts Institute of Technology has compiled easily applicable tables, based on the observations of sliding movements which have occurred in the United States.

We also publish in this Bulletin the application of these principles by John P. SLEE, B. Sc. (Eng.), Civil Engineer's Department, Western Region, British Railways, in the study of an earth slide which occurred at Sonning cutting on the Paddington to Reading line.

In conclusion, we would wish that Mr. LAZARD could soon be able to publish his views on the widening of the conception of safety by introducing notions of statistics and probabilities, in keeping with the original ideas of the French « probability » school of thought and, in particular, of Mr. Robert LÉVI, Director of Fixed Installations, French National Railways.

[625 .12 (42)]

Landslides at Twyford and Sonning on the Paddington-Reading line.

The *Railway Gazette*, in the issues for the 22nd and 29th May 1953, published two articles about landslides which occurred on railway cuttings on the Paddington-Reading line at Twyford and Sonning.

The geological formations are practically identical in both cases : the clay topsoil slid away from the thin sandy underlayer which at Twyford had suffered particularly severe hydrostatic underpressure due to the heavy rains of 1951.

The remedies applied, practically the same in both cases, consisted in anchoring the shifting ground to a stable clay formation under the layer of sand by means of buttresses of stones acting both as a drain and a reinforcement (the well known De Sazilly method) and reducing the water pressure in the sand by carrying away the water to the drains in the cutting.

In both cases, at the top of the cutting, the classical vertical crack was observed before the landslide occurred.

The two very instructive articles are from the pen of Mr. John P. SLEE, B. Sc. (Eng), Civil Engineer's Department, Western Region.

We thought it of value to publish in extenso the study he made of the landslide at Sonning.

The method followed is similar to that used in Sweden and suggested by Professor FELLENIUS.

Repairing unusual slips on Western Region main line,

by John P. SLÈE, B.Sc. (Eng.), Civil Engineer's Department, Western Region.

(From *The Railway Gazette*, May 29, 1953).

The slip is assumed to be circular from ground level on the bank to a point vertically beneath the centre of the slip circle; as seen in figure 1, this is a close approximation. It is the stability of the mass enclosed by the radii of this circle under discussion.

The forces acting on it are : the out-of-balance moment of the mass; the active resistance of the ballast (this value is taken as the ballast is likely to be disturbed during the course of day-to-day maintenance); the passive resistance of the clay between the sand layer and the ballast; and the resistance of the sand layer as a result of its shear strength.

These are the forces that applied at the time of the first slipping. It was not until the subgrade was disturbed by the slip that its strength was sufficiently reduced to carry the slip circle through it. At a later stage the slip undoubtedly changed its character, as was shown by the change in the movement of the down road. The path of the slip circle through the subgrade is shown by a dotted line.

All these forces can be considered in

terms of their moment about the centre of the slip circle. They have all been calculated for a strip a foot wide and the results are shown below :

1) The out-of-balance moment M_B calculated using unit weight of clay as 136 lb./cu. ft.

$$M_B = 4\,973\,000 \text{ lb. ft.}$$

2) The passive resistance of the clay subgrade is calculated, as demonstrated for Twyford, from the formulae

$$P_{E1} = \frac{H}{2}\left[\gamma H \tan^2\left(45 + \frac{\Phi}{2}\right) + 4\,c \tan\left(45 + \frac{\Phi}{2}\right)\right]$$

P_{E1} acting at 53 ft. from the slip circle centre = 50 450 lb.

$$P_{E2} = W_B\, h_B\, H.$$

P_{E2} acting at 52 ft. from the slip circle centre = 2 880 lb.

The stabilising moment of the passive earth pressure equals :

$$M_P = 2\,820\,000 \text{ lb. ft.}$$

3) As was found in the calculations for the Twyford slip, the active earth pressure was negligible.

4) To give a factor of safety of unity the moment of the shear strength of the sand layer about the slip circle centre equals M_S.

$$M_S = M_B - M_P$$

$$M_S = 2\,153\,000 \text{ lb. ft.}$$

The length of the sand layer = 76 ft. and its radius 57.4 ft.

The average shear strength at unit safety factor equals :

494 lb. per sq. ft.

The satisfactory determination of the shear strength of the curved layer is very difficult, but it can be safely assumed that at some stage it reached this value and failure

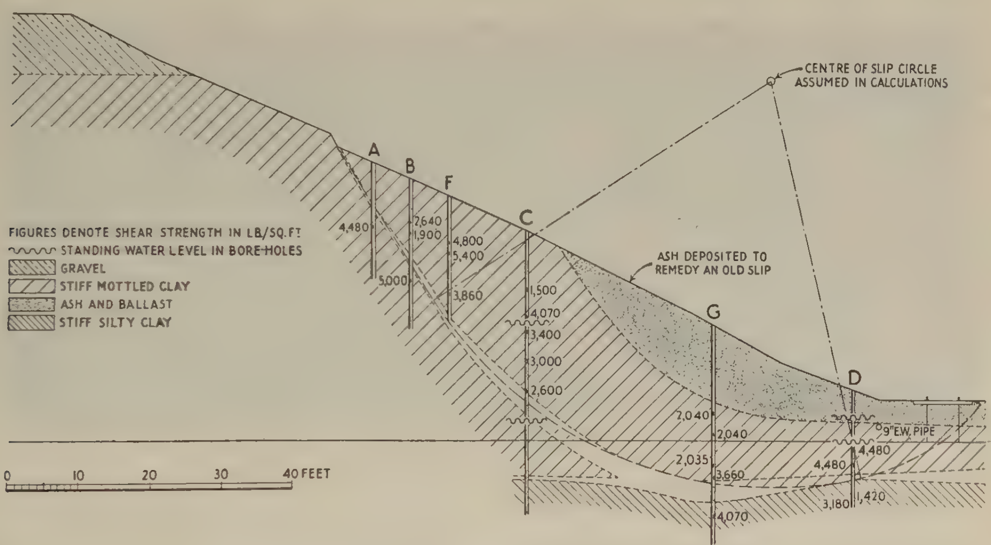
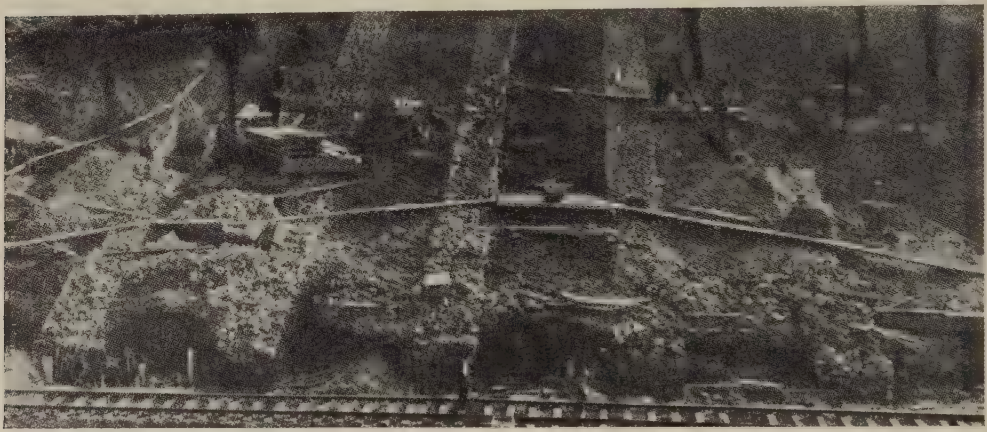


Fig. 1. — Cross-section through the worst part of the slip in Sonning Cutting.

resulted. It was assumed that this value could fall to 350 lb./ft.^{2} and if this happened the force to be provided by the remedial works was 5 tons/ft. run at the sand level.

In repairing both slips the same principle was applied. It is proposed to describe the works at Sonning first, as they were much greater in extent and cost, and describe those at Twyford by referring to differences between them and those at Sonning.



Sonning Cutting, remedial work nearing completion.

The cause of the slip at Sonning was known exactly and, paradoxically, this meant that the ways of dealing with it were numerous. The method decided on will be described and other methods, with the reasons for their rejection, touched on briefly afterwards. The object of this method was to fix the slipping mass to the solid stratum beneath.

Buttressing.

The means used were the construction of seven buttresses or keys, sufficient in size to withstand the force anticipated in the calculations given previously.

The buttresses were made by digging holes 12 ft. by 7 ft. in plan, in a pattern as shown in figure 2 to allow for easy working without the fear of weakening the cutting too much. They were carried through the sand layer and into the silty clay stratum as shown. This work was carried out while the slip was in progress, and heavy timbering was used, a necessary precaution as was shown by the considerable movement that took place. Compressed air clay spades and Neals cranes were the only power equipment used, and work progressed without difficulty until the sand layer was reached and pumping had to be started. At this stage, trouble was caused when the pumps, carrying a quantity of sand away with the water, caused large holes to be formed in the sand layer around the excavation.

The timbering was carried successfully through into the silty clay layer, however, and the flow of sand largely cut off. These holes were filled in when the stone backfill was placed. The stone backfill was laid in a matrix of 1 : 12 concrete except in the first two holes to be completed. No concrete was used in these and a 12-in. dia. earthenware pipe, was placed vertically in the middle of them was used as a sump from which the general water level could be kept to a minimum, thereby increasing the strength of the sand layer. In addition,

1 1/2 in. gas pipes were placed so that grouting of the stone could be carried out at a later date.

It will be seen from figure 2 that successive 12 ft. by 7 ft. holes were constructed, one behind the other, to form the buttresses; the maximum and minimum depths of excavation were 39 ft. and 21 ft. After the first of these had been constructed, a noticeable decrease in movement of the track was apparent. However, the cutting slope began to belly out.

To facilitate the construction of the second series of 12 ft. by 7 ft. holes, the timbering on the high side of the first series had been left in place. When these were again exposed by the excavation it was found that they had moved from a vertical position to such an extent that over a length of 26 ft. they were 2 ft. 10 in. out of true. These developments, which were welcomed in view of the minimising of the threat to the running lines, illustrated that a secondary slip was occurring above the level of the tracks. This was anticipated by the calculations, which showed that the most likely type of failure to which the buttresses were subject was overturning.

To transfer the thrust from the slipping mass on to the buttresses, counterforts 7 ft. deep will be constructed running up the bank. These also act as drains. The water which collects in the gravel stratum will be tapped by shallow drains connected to the counterforts. It is intended in this way to prevent surface water running into the exposed top of the sand layer; in addition this will be covered with clay and punned down.

This will be reinstated where the old earthenware pipes have been broken up by the slip. The water will be collected at the level of the top of the grouted part of the buttresses and then run into the centre cess drain at about 6 ft. 6 in. below rail level. The new slope will be compounded, 1 : 3 over the slipping area and 1 : 1.5 over the unaffected part. This in itself reduces the out of balance force by 23 per cent and will add to the stability of the completed work.

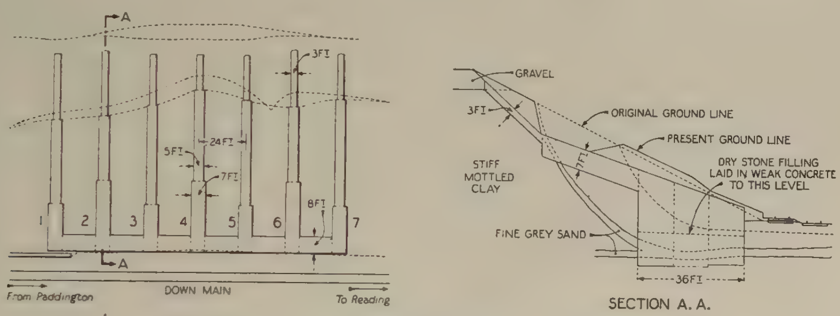


Fig. 2. — Diagram of slip repairs at Sonning.

Calculations of factor of safety.

Out of balance moment = 3 819 000 lb. ft. This is to be counter-balanced by the strength of the sand layer (350 lb. per sq. ft.) = 1 500 000 lb. ft. The passive earth pressure is assumed to act in reducing the out of balance moment between the counterforts only. The pressure on the front of the buttress is used in considering their stability.

Stabilising moment of passive earth pressure $= \left(\dfrac{24 - 7}{24}\right) \times 2\,820\,000 =$ 2 000 000 lb. ft. ∴ Unbalanced moment to be withstood by the buttresses equals 319 000 lb. ft.

The criterion for deciding the safety factor is the overturning of the concreted portion of the buttress. The forces acting

are shown in figure 3. The calculation is carried out for the smaller of the two types of buttress which is 24 ft. by 7 ft. in plan. The whole unbalanced moment is assumed to act through the counterforts and resolves into the force F_H and a force F_V which, acting through the centre of the slip circle, has no moment. F_H acts at a perpendicular distance of 51 ft. from the centre of the slip circle.

$$F_H = \frac{319\ 000}{51 \times 2\ 240} \times 24 = 67 \text{ tons.}$$

U upthrust on the buttress due to the water surrounding it = 47 tons.

P_{E1} and P_{E2}. The presence of the sand layer is ignored in calculating these values and they can be found in the same way as before.

P_{E1} = 1 890 lb. = 0.84 tons.

P_{E2} = 244 300 lb. = 105 tons.

Using the forces given above and the dimensions shown, figure 3, it is possible to take moments about the toe or heel of the buttress. From this the reaction

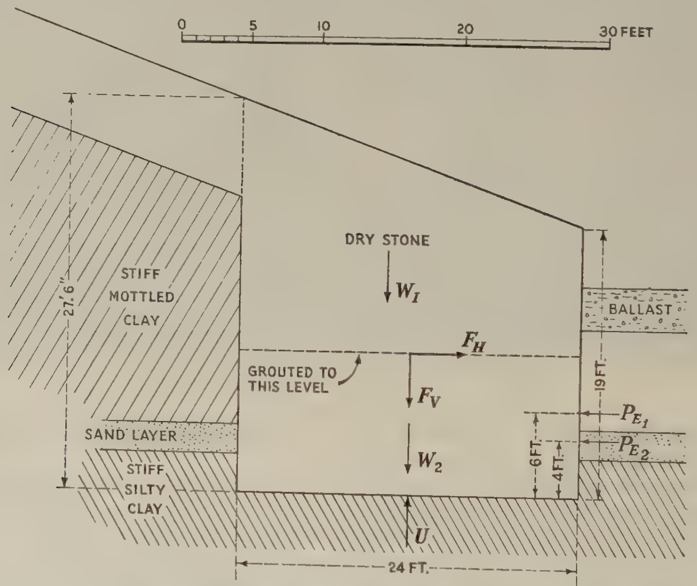


Fig. 3. — Diagram of forces acting on the concreted portion of the buttress.

F_V. — Resolving F_H with F_V and assuming the angle of shearing resistance of the drystone to be 60 deg. then,

$$F_V = F_H \tan\left(\frac{2}{3} \times 60^o\right) = 56 \text{ tons.}$$

W_1 is the weight of the drystone = 119 tons.

W_2 is the weight of the concreted part of the buttress = 105 tons.

on the base can be found and hence the greatest stress on the base is 1.4 tons per sq. ft. The ultimate bearing stress by experiment = 3.5 tons per sq. ft. Thus the factor of safety for the smaller type buttress equals 2.5. The factor of safety for the larger type of buttress can be calculated in the same way and equals 3.1. Thus the overall safety factor lies somewhere between these values, and these are considered to be satisfactory limits.

When remedial measures were first called for, the overriding factor to be considered was which method would have the quickest effect in arresting the movement of the main line. The easiest emergency measure to apply was the construction of buttresses. The job could begin without plant; timber and hand tools and labour skilled in this type of work were readily available. In the first instance, the construction of buttresses was the obvious course, and two holes were started on March 10, 1953. At the same time calculations of efficiency in terms of safety factor for assumed conditions and estimates of cost were made.

Twyford slip, Cutting after completion of remedial work.

Reinforced concrete wall.

One scheme involved the construction of a reinforced concrete retaining wall founded in the stable stratum below the sand layer and extending the length of the slipping area. The thrust would be transferred from the clay mass to the wall and thence to the stable stratum. The cost for a similar factor of safety was about the same; this solution was not used because the skilled labour required for reinforced concrete construction was not readily available.

Compounding of slope.

A satisfactory factor of safety could be obtained by compounding the existing slope of 1 : 2 to be 1 : 4 over the slip and 1 : 1 1/2 above it. This would have the effect of moving the edge of the slope back a maximum of 8 ft. which would still not involve any movement of the boundary fence. This scheme envisaged the use of a mechanical excavator operating from the stable portion above the slope. This idea was regretfully abandoned (it involved a saving in cost of about 50 per cent) as it was felt that traffic might be endangered.

The prime cause of the slip at Twyford was the hydrostatic head which had collected in the sand lens. This was reduced to an economic minimum by the use of 6-in. dia. cast-iron pipes, with perforated ends, which were built into the buttresses, the perforations being at the level of the sand lens. The pipes terminated slightly above the level of the drain in the centre cess, that is 12 ft. above the sand lens. The buttresses

are then to be connected by earthenware pipes to the centre cess drain. The hydrostatic head is limited by these measures to 12 ft.

A slip plane having formed, and the effective strength of the clay being greatly reduced, it was necessary also to provide another stabilising factor which also took the form of buttresses and counterforts. As the original slope was not so great as at Sonning the remedial measures were not so extensive. The buttresses measured 24 ft. by 7 ft. and were spaced at 21 ft. centres, and the counterforts will be made to drain the gravel layer directly. The buttresses were constructed in dry stone with no concrete matrix.

This article shows how a new cutting may be threatened by the existence of permeable strata which are below the limit of excavation. If a preliminary investigation has been carried out and the presence of a permeable stratum found, the correct slope of the new cutting can be found using the calculation methods used in this article.

The author is indebted to Mr. M. G. R. SMITH, Civil Engineer, Western Region, for permission to publish this report; Mr. P. PROTOPAPADAKIS, Development Assistant, for his advice and guidance; and Mr. D. L. BARTLETT.

BIBIOGRAPHY

K. TERZAGHI and R. B. PECK, 1948, "Soil Mechanics in Engineering Practice", pages 366 to 370.

H. R. REYNOLDS and P. PROTOPAPADAKIS, 1948, "Practical Problems in Soil Mechanics", page 87.

* * *

A third landslide threatened an important set of lines near Uxbridge Station, London Transport, but was averted in rather an original way (*Railway Gazette*, 31st December 1954).

The lines are on a lower level than the adjoining ground which is partly built over.

On one side, the 45 ft. deep bank of the cutting is supported by a wall of varying height up to 30 ft. which is slightly curved.

Before the cylindrical slide of the mass of earth including the wall and the track over some 200 ft. the usual signs were noted, namely longitudinal cracks in a cross road above the cutting.

By means of a drag line the earth of the cutting was removed, chiefly soft yellow clay. Concrete supports were run into hollows made under the track supported by the existing walls or on new concrete supports.

The tracks were then raised 4 ft. above their original level by unloading stones to weigh down the bulge which tended to occur.

The use of supports under the track and the raising up of the track, an original solution, made it possible to prevent any further landslide.

J. DUBUS.

[385. (09 .2]

OBITUARY.

Lt.-Colonel Sir Alan MOUNT, C.B., C.B.E.,

Chief Inspecting Officer of Railways, Ministry of Transport, 1929-1949.
Former Member of the Permanent Commission of the International Railway Congress Association.

We regret to record the death, on August 10, of our former Colleague, Lt.-Colonel Sir Alan Mount, Chief Inspecting Officer of Railways from 1929-1949.

Sir Alan Mount was born in 1881 and educated at Bradfield and at the Royal Indian Engineering College, Coopers Hill, of which he was made a Fellow. He was commissioned in the Royal Engineers in 1902, and spent two years at the School of Military Engineering, Chatham, also gaining a year's practical experience in the Locomotive Department of the Midland Railway. In 1905 he proceeded to India, and was appointed as Assistant Engineer on the North Western Railway. In 1911-12 he was placed on deputation as Assistant Engineer in charge of the Delhi Durbar light railways, and was awarded the Kaiser-i-Hind Medal for his services in that connection. Later in 1912 he was attached to the Town Planning Committee of New Delhi as Executive Engineer, and prepared the scheme for the terminal arrangements there.

On the outbreak of war in 1914 he reverted to military duty and left for France with the Lahore Division. Later he joined the Railway Directorate on the Western Front and became an assistant director of railway construction at G. H. Q. with the temporary rank of Lt.-Colonel; in 1917 he was appointed Deputy Chief Construction Engineer for broad-gauge railways under the Director General of Transportation. He was mentioned in despatches four times, and promoted Brevet-Major in 1916. In 1917 he received from the French President the Cross of the Legion of Honour; he was also made a C. B. E. for his services in the field.

On January 1, 1920, he was appointed an Inspecting Officer of Railways in the Ministry of Transport, becoming Chief Inspecting Officer in 1929, having retired from the Royal Engineers in 1922. He was

made a C. B. in 1931, and received the honour of knighthood in 1941 for services on the Indian Pacific Locomotive Committee, of which he was Chairman.

Sir Alan MOUNT was also Chairman of the Railway Workshops Capacity Committee 1940, and a member of the Railway (London Plan) Committee, 1944. He retired from the Ministry of Transport in 1949, and was appointed Consultant to the Railway Executive on safety measures for British Railways.

(The above biography is an extract from *The Railway Gazette.*)

Sir Alan MOUNT was elected member of the Permanent Commission of the International Railway Congress Association in February 1946. His participation to the works of our Association dates back to the London Congress of 1925, where he was a delegate of the British Government. He took also part as a delegate of the Ministry of Transport to the Madrid (1930), Cairo (1933) and Paris (1937) Congresses. At these two last Sessions, he actively and remarkably participated to the discussions of the questions relating to the safety of operation and particularly concerning the automatic train control and train stop. After the war, Sir Alan MOUNT represented also the British Government at the Lucerne Congress in 1947. He resigned from the Permanent Commission in 1951.

All his colleagues will remember him for his kindness and as a devoted friend of our Association.

We wish to convey our very sincerest sympathy to his family.

The Executive Committee.

NEW BOOKS AND PUBLICATIONS.

[**656** (02]

BOURGEOIS René, Chef Adjoint de la Direction Commerciale de la Société Nationale des Chemins de fer français. — **Exploitation commerciale des Chemins de fer français.** *(The commercial operation of the French Railways.)* — One volume (6 1/4 × 9 7/8 inches) of 460 pages, illustrated, with 36 plates and 4 folders. — 1955, Paris, Léon Eyrolles, Publishers, 61, Boulevard Saint-Germain (Price : 2 900 French fr.).

The commercial operation of a railway undertaking, especially such an extensive and varied railway system as the S.N.C.F., is of the foremost importance at the present time.

This has the mission of assuring relations with its clients, with the Public Authorities, with other railways, with other transport undertakings, even with its competitors. It is responsible for defining the nature and quality of the services offered, as well as the rates and other transport conditions.

The subject chosen by Mr. Bourgeois was therefore extremely vast. Even discarding all but the essential ideas, and the principles of the rating system, and limiting his study to the French Railways in the year 1954, a very large book was necessary to cover the essential activities of the commercial department.

The plan of the book covers three main parts.

The first, which is in some way preparatory, deals with the « Factors of the commercial problem for the railway ». These factors are contained first of all in the regulations imposed on the railway and the intervention of the Public Powers. Others, not the least important, are dictated by economic circumstances. Finally, there are the new arrivals, already not so new, competition and co-ordination.

The two other parts deal respectively with freight traffic and passenger traffic. It is a study of the solutions brought to many problems in these two fields, which might be stated in these general terms : to give the best possible service within the limits imposed by the finances of the railway.

As it is impossible to cover so large a field in a few lines, we will merely report the main points dealt with in the book.

The first part contains an analysis of the specifications and financial agreements with the State. The idea of costs is dealt with in order to show their effects upon the rates.

Its many obligations and small amount of liberty were amongst the circumstances which favoured competition. This led to a great many reactions, especially the various co-ordination measures.

In the case of freight traffic, the question of carrying out the transport in its various forms serves as a preliminary to a very detailed study of the rates. These, owing to their diversity, tend to meet as far as possible all the requirements of clients, increase the usefulness of the railway, and encourage the receipts. One chapter deals with the links with other methods of transport and railways abroad.

In the case of passenger traffic, the author shows the great variation there is in time and space, and looks into the causes of the deficit. Differential rates are due to special conditions, but also are a result of the railway's preoccupation with meeting the requirements of as large a clientele as possible. A chapter

dealing with international traffic shows its importance and characteristics, and the progress made thanks to international tariffs.

The fourth part describes the commercial organisation of the S. N. C. F. and how it works to stimulate the activity of the railway.

In his conclusion, the author insists upon the commercial and industrial character of the railway without forgetting that it is a public service which imposes upon it a burden unknown to ordinary commercial undertakings.

E. M.

MONTHLY BIBLIOGRAPHY OF RAILWAYS (1)

PUBLISHED UNDER THE SUPERVISION OF

P. GHILAIN,

General Secretary of the Permanent Commission of the International Railway Congress Association.

(NOVEMBER 1955)

[**016.** 385 (02]

I. — BOOKS.

In French.

1955 **385**
L'Année ferroviaire 1955.
Paris, Plon, éditeur, 8, rue Garancière. Un volume in-8° de IV-224 pages, avec gravures et planches hors-texte. (Prix : 690 fr. fr.)

1955 **6** (4)
ARMAND (L.).
Coopération économique intra-européenne dans le domaine de l'énergie.
Paris (XVI^e^), Organisation Européenne de Coopération Economique, 2, rue André-Pascal. Une brochure (21 × 27 cm) de 112 pages.

1955 **69** (02
GUILLOUX (P.).
Notions sur les ouvrages de travaux publics.
Paris, Eyrolles, éditeur. Un volume (21 × 27 cm) de 264 pages, avec 570 figures et une planche hors-texte. (Prix : 2 900 fr. fr.)

1955 **621** .83
HENRY (M.) & POURET (H.).
Taillage, rasage et rectification des engrenages.
Paris (V^e^), Eyrolles, éditeur, 61, boulevard Saint-Germain. Un volume (16.5 × 25 cm) de 184 pages, avec 195 figures et 16 planches hors-texte. (Prix : relié toile, 1 480 fr. fr.)

1955 **531**
Journées Internationales de l'Usinage (Liège, 27 au 30 avril 1955). Première partie. *Revue Universelle des Mines*, n° 9, septembre 1955.
Liège, *Revue Universelle des Mines*, 22, rue Forgeur. Un fascicule (20.5 × 30 cm) de 66 pages, avec illustrations. (Prix : 100 fr. belges.)

1955 **665** .882
LEROY (A.), EVRARD (M.) & d'HERBEMONT (G.).
La pratique du soudage oxyacétylénique et des techniques connexes.
Paris, Publication de la Soudure Autogène, 39, rue d'Amsterdam. Un volume (13 × 21 cm) de 344 pages, avec 210 figures et tableaux. (Prix : 600 fr. fr.)

In German.

1955 **656** .235
Dr. rer. pol. Richard COUVÉ.
Die Eisenbahngüterabfertigung. Teil II. 9. neubearbeitete und erweiterte Auflage.
Frankfurt (Main), Verkehrswissenschaftliche Lehrmittelgesellschaft m.b.H. VIII-212 Seiten. (Preis : DM 9.50.)

1955 **656** .254
Elsners Taschenbuch für den fernmeldetechnischen Eisenbahndienst 1955. 5. Jahrgang.
Frankfurt (Main), Dr. Arthur Tetzlaff-Verlag. 222 Seiten, Format DIN A6. (Preis : karton. DM 4.20; für Eisenbahner DM 3.80.)

1955 **385**
Jahrbuch des Eisenbahnwesens 1955. Herausgegeben von Berthold STUMPF unter Mitarbeit namhafter deutscher und ausländischer Fachleute.
Darmstadt, Carl Röhrig-Verlag, Stephanstrasse, 8. 203 Seiten Text mit vielen Bildern, Zeichnungen und Tafeln, DIN A 4. (Preis : DM 7.20.)

1955 **691**
Dr.-Ing. W. SÄGER.
Der Einfluss des Kriechens und Schwindens in Spannbetonkonstruktionen.
Düsseldorf, Werner-Verlag. 175 Seiten mit 25 Bildern im Text.

1952 **62** (01
Tafeln zur Ermittlung der Brinellhärte. Neubearbeitet von N. LUDWIG.
Berlin W 15 und Köln, Beuth-Vertrieb GmbH. DIN A 5, 31 Seiten. (Preis : DM 3.—.)

In English.

1955 **656** .23
BUNNELL (E.H.).
Railroad accounting and statistics
One volume (6 ½ × 9 ½ inches) of 272 pages, 50 illustrations, fully indexed.
Chicago : Watson Publications, Publishers of Modern Railroads, 201, N. Wells Street, 6 Ill. (Price : 6 $.)

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(1) The numbers placed over the title of each book are those of the decimal classification proposed by the Railway Congress conjointly with the Office Bibliographique International, of Brussels, (See « Bibliographical Decimal Classification as applied to Railway Science », by L. WEISSENBRUCH, in the number for November 1897, of the *Bulletin of the International Railway Congress*, p. 1509.)

1955 621 .431.72
Diesel engines. Principles and practice.
Edited by C.C, POUNDER.
London : George Newnes, Ltd., Tower House, Southampton Street, Strand, W.C. 2. (Price : 84 s.)

1954 **669**
LIPS (E.M.H.).
Engineering metallurgy.
One volume of 230 pages, figs and tables.
Eindhoven : Phillips' Technical Library. (Price : 32 s. 6 d. stg.)

1955 **621** .33
MATHEWS (P.).
Protective current transformers and circuits.
Chapman and Hall, Ltd., 37, Essex-street, W.C. 2. (Price : 36 s.)

1955 **385** (09
MORGAN (BRYAN).
The end of the line.
London : Cleaver Hume Press, Ltd., 31, Wright's Lane, Kensington, W. 8. (Price : 25 s.)

1955 **385** .5
NOCK (O.S.).
The railway engineers.
One volume (8 ½ × 5 ½ in.) of 256 pages, illustrated.
London : BT. Batsford Limited, 4, Fitzhardinge Street, Portman Square, W. 1. (Price : 18 s.)

1955 **385** (09 (42)
NOCK (O.S.).
Fifty years of Western Express running.
Bristol : Edward Everard, Ltd., 38, Broad Street. (Price : 25 s.)

1955 **621** .431.7
PARKINSON (T.H.).
Diesel maintenance. *Fourth edition.*
Edited by Donald H. Smith.
London : Iliffe and Sons, Limited, Dorset House Stamford Street, S.E. 1. (Price : 12 s. 6 d. net.)

1955 **656** .
WEBB (W.L.).
Railroad construction. *Ninth edition.*
One volume ($4\,^3/_4 \times 7\,^3/_8$ inches) of 226 pages illustrated.
London : Chapman and Hall, Ltd., 37, Essex Street W.C. 2.

In Italian.

1955 **65**
Prof. Ing. M. MATERNINI.
Trasporti. — Compendio delle principali question tecniche ed economiche dei vari mezzi di trasporto.
Trieste, Libreria Internazionale L. Cappelli. 350 pagin con 140 illustrazioni. (Prezzo : L. 2 800.)

1955 **721** .
SANSONI (R.).
Pali e fondazioni su pali.
Milano, Ulrico Hoepli, edittore, 290 pagine, 19 figure e 40 tavole. (Prezzo : L. 2 500.—.)

[**016.** 385 (05]

II. — PERIODICALS.

In French.

Annales des ponts et chaussées. (Paris.)

1955 **62** (01
Annales des ponts et chaussées, juillet-août, p. 403.
BONNEAU (L.). — Problèmes **d'élasticité à trois dimensions.** L'équation de Lagrange et les plaques rectangulaires. (10 000 mots, tableaux & fig.)

1955 **625** .5
Annales des ponts et chaussées, juillet-août, p. 483.
LEHANNEUR (L.). — Observations sur deux mémoires de M. Nicolas concernant **l'équilibre des câbles de téléphériques.** (4 000 mots & fig.)

Bulletin technique de la Suisse Romande. (Lausanne.)

1955 **669** .1
Bulletin techn. de la Suisse romande, 20 août, p. 257.
AUER (R.C.). — **Aciers rapides et aciers indéformables sulfinisés.** (1 500 mots & fig.)

Bulletin des Transports Internationaux par Chemins de fer. (Berne.)

1955 **347** .76
Bull. des transp. intern. par ch. de fer, août, p. 21
de NANASSY (B.). — **La conclusion du contrat transport des voyageurs par chemins de fer.** (4 000 mots

---

1955 **385** .113 (49
Bull. des transp. intern. par ch. de fer, août, p. 24
Rapport des Chemins de fer fédéraux sur la gestio et les comptes de l'année 1954. (3 000 mots.)

Economie et Technique des Transports. (Zurich.)

1954 **625** .232 (4
Economie et Technique des Transports, n° 7-12, juille décembre, p. 77.
JOHNSTON (J.J.). — **C.I.É. Buffet cars.** (2 000 mo & fig.)

1954 **656** .221
onomie et Technique des Transports, nº 7-12, juillet-décembre, p. 98.
Prof. W. BÄSELER. — Die zukünftige **Bewältigung** n **Steilrampen.** (3 000 mots & fig.)

1955 **625** .212
onomie et Technique des Transports, nº 1-6 p. 49.
Appareil pour mesurer le diamètre aux bandages des ues. (500 mots & fig.)

1955 **62** (01 : **625** .5
onomie et Technique des Transports, nº 1-6, p. 50.
BAUD (R.V.) & MEYER (J.). — **Magnetische** rüfung **von Kabeln** von Drahtseilbahnen, Kabelkranen d dergl. (2 000 mots & fig.)

Electricité. (Paris.)

1955 **62** (08
ectricité, mars, p. 74.
ROBERT (O). — **Etude générale des systèmes d'unités** **des normes scientifiques françaises et internationales.** *uite)*. (1 000 mots & tableaux.)

1955 **621** .31
ectricité, avril, p. 91; mai, p. 137; juin, p. 162.
PASSAQUIN (J.). — Résultats d'essais de **disjoncteurs** **air comprimé à haute tension.** (9 000 mots & fig.)

Génie Civil. (Paris.)

1955 **624** .5
énie Civil, nº 3401, 15 juillet, p. 265.
CHADENSON (L.). — L'importance des problèmes rodynamiques dans la conception des grands **ponts** **spendus.** (3 000 mots & fig.)

1955 **621** .133.1
énie Civil, nº 3402-3403, 1er-15 août, p. 301.
KIMPFLIN (G.). — Le **Quatrième Congrès mondial** **Pétrole.** (Rome, 6-15 juin 1955). (2 000 mots & cartes.)

1955 **624** .8 (44)
énie Civil, nº 3402-3403, 1er-15 août, p. 304.
La **reconstruction de la travée tournante du viaduc** **Caronte** (Bouches du Rhône). (1 500 mots & fig.)

1955 **625** .245 : **621** .33 (44)
énie Civil, nº 3402-3403, 1er-15 août, p. 308.
La **voiture électrotechnique** de la Société Nationale des hemins de fer français. (1 000 mots & fig.)

'Industrie des Voies ferrées & des Transports automobiles. (Paris.)

1955 **625** .6 (494)
'Industrie des Voies Ferrées et des Transports Automobiles, juillet-août, p. 103.
DELGOFFE. — Le **Chemin de fer d'Yverdon à** **ainte-Croix.** (1 500 mots & fig.)

Revue Générale des Chemins de fer. (Paris.)

1955 **621** .33 (44)
Revue Générale des Chemins de fer, juillet, p. 429.
Journées d'Information de Lille (11-14 mai 1955). **Discours d'ouverture** de M. Louis ARMAND. (4 000 mots.)

1955 **621** .33
Revue Générale des Chemins de fer, juillet, p. 436.
PARODI (H.). — **L'évolution des installations de traction électrique.** (2 000 mots.)

1955 **621** .33
Revue Générale des Chemins de fer, juillet, p. 441.
GARREAU. — **Bilans comparés des différents systèmes de traction électrique.** (15 000 mots, carte et tableaux.)

1955 **621** .332
Revue Générale des Chemins de fer, juillet, p. 467.
GASTINE. — Le Réseau public d'énergie pour l'**alimentation de la traction électrique.** (3 000 mots & fig.)

1955 **621** .331
Revue Générale des Chemins de fer, juillet, p. 478.
BERNARD. — La position et la constitution des **sous-stations en monophasé 50 Hz.** (6 000 mots & fig.)

1955 **621** .332
Revue Générale des Chemins de fer, juillet, p. 489.
WALTER. — Les **caténaires.** Quelques problèmes de construction et d'entretien. (4 000 mots & fig.)

1955 **621** .332
Revue Générale des Chemins de fer, juillet, p. 497.
CRÉPET (A.). — Les **caténaires.** Quelques types d'installations et d'appareils. (10 000 mots & fig.)

1955 **621** .332 (44)
Revue Générale des Chemins de fer, juillet, p. 521.
RAOULT. — **Alimentation et exploitation électrique de la caténaire** Valenciennes-Thionville. (5 000 mots & fig.)

1955 **621** .332 (44) & **656** .254 (44)
Revue Générale des Chemins de fer, juillet, p. 531.
COLLET (L.J.). — **Coexistence des lignes électrifiées en courant alternatif et des lignes de télécommunications.** (2 000 mots.)

1955 **621** .332 (44) & **656** .254 (44)
Revue Générale des Chemins de fer, juillet, p. 536 et 546.
WALTER. — **Principes de l'équipement des installations de télécommunications et de signalisation.** (5 000 mots & fig.)
TROGNEUX. — Etudes et essais effectués par la S.N.C.F. sur ses **installations de télécommunications et de signalisation.** (8 000 mots & fig.)

1955 **621** .335 (44)
Revue Générale des Chemins de fer, juillet, p. 568, 582, 595 et 619.
NOUVION (F.). — Considérations générales sur les **matériels moteurs monophasés actuels.** (6 000 mots & fig.)

NOUVION (F.). — L'évolution prochaine des **locomotives monophasées à 50 Hz.** (5 000 mots & fig.)
TESSIER (M.). — Le **matériel moteur à « redresseurs ».** (12 000 mots & fig.)
GAIDE. — Les **locomotives à moteurs à collecteur à 50 Hz.** (6 000 mots & fig.)
BOILLOT. — Les **locomotives à groupe convertisseur tournant.** (4 000 mots & fig.)

1955 **621** .33 (44)
Revue Générale des Chemins de fer, juillet, p. 652.
Enseignements tirés des Journées de Lille par la S.N.C.F. (1 000 mots.)

Les Transports Publics. (Berne.)

1955 **656** (494)
Les Transports publics, septembre, p. 3.
Les **moyens de transport** en Suisse et leurs **problèmes.** (1 000 mots.)

1955 **656** .254
Les Transports publics, septembre, p. 5.
Du nouveau à propos de la **signalisation automatique des passages à niveau.** (1 200 mots.)

Travaux. (Paris.)

1955 **625** .12
Travaux, septembre, p. 707.
LAZARD (A.). — Nouvelles remarques sur le **calcul de la stabilité des talus en terre.** (7 000 mots & fig.)

In German.

Die Bundesbahn. (Darmstadt und Köln.)

1955 **385** .57 (43)
Die Bundesbahn, Nr. 11, Juni, S. 449.
STIELER (C.). — Die **Ausbildung und Prüfung der Schweisser** der Deutschen Bundesbahn. (1 500 Wörter & Abb.)

1955 **385** (43)
Die Bundesbahn, Nr. 11, Juni, S. 463.
Weitere **Rationalisierungserfolge** der Deutschen Bundesbahn bei den Dienststellen des Aussendienstes. (2 000 Wörter.)

Deutsche Eisenbahntechnik. (Berlin.)

1955 **656** .212
Deutsche Eisenbahntechnik, Juni, S. 223.
HOCHSTEINER (O.) & POTTHOFF (G.). — **Sammelbahnhöfe.** (5 000 Wörter & Abb.)

1955 **621** .431.72 (43)
Deutsche Eisenbahntechnik, Juni, S. 231.
WUNDRACK (A.). — Die **vierteiligen diesel-mechanischen Triebwagenzüge** der Deutschen Reichsbahn. (3 000 Wörter & Abb.)

1955 **656** .221
Deutsche Eisenbahntechnik, Juni, S. 237.
ECKHARDT (F.W.). — Der **Eigenwiderstand von Dampflokomotiven.** (2 000 Wörter, Tafeln & Abb.)

Der Eisenbahningenieur. (Frankfurt a. Main.)

1955 **721** .4
Der Eisenbahningenieur, Juni, S. 148.
HENNINGS (W.). — **Stirnmauerausbildung bei Gewölben.** (1 000 Wörter & Abb.)

1955 **625** .142.4
Der Eisenbahningenieur Juni, S. 150.
HÄRTEL (H.). — **Verlegen der Betonschwellen,** Bestimmungen der Umbaulängen. (1 200 Wörter.)

1955 **625** .122
Der Eisenbahningenieur, Juni, S. 151.
LAUN (K.). — **Sicherung von Felswänden.** (1 500 Wörter & Abb.)

1955 **625** .1 : **625** .162
Der Eisenbahningenieur, Juni, S. 155.
VOLBACH (J.). — **Wiederaufbau der Unterführung** Bonner Strasse in Siegburg. (1 500 Wörter & Abb.)

1955 **621** .31 (43)
Der Eisenbahningenieur, Juni, S. 158.
HENRICI (G.). — Die **Stromtarife** bei der Deutschen Bundesbahn. (3 000 Wörter.)

1955 **621** .431.72 (43)
Der Eisenbahningenieur, Juni, S. 161.
LAND (W.). — Die **Henschel-General Motors-Diesellokomotive.** (1 000 Wörter & Abb.)

1955 **625** .3
Der Eisenbahningenieur, Juni, S. 164.
PLAG. — **Bringt die ALWEG-Bahn eine Teillösung der grossen Verkehrsprobleme?** (1 000 Wörter & Abb.)

1955 **625** .111
Die Eisenbahningenieur, Juni, S. 166.
FENDESACK (H.). — Wirtschaftliches Arbeiten im **Vermessungsdienst** durch geeignete Hilfsmittel und Methoden. (3 000 Wörter.)

E.T.R. Eisenbahntechnische Rundschau. (Köln-Darmstadt.)

1955 **656** .254 (43)
Eisenbahntechnische Rundschau, August, S. 325.
STROH (A.) & KLEINBAUER (K.). — Betriebliche und signaltechnische Erfahrungen mit der **Fernsteuerung** auf der Strecke Nürnberg-Regensburg. (8 000 Wörter & Abb.)

1955 **625** .142.4
Eisenbahntechnische Rundschau, August, S. 343.
MEIER (H.). — Betrachtungen zur Problematik der **Betonschwelle.** (6 000 Wörter & Abb.)

1955 **621** .132.8
Eisenbahntechnische Rundschau, August, S. 361.
AIGNER (J.) & DISTELBARTH (W.). — Eine **Zahnradlokomotive grosser Leistung für Meterspur.** (5 000 Wörter & Abb.)

1955 **625** .143.3
Eisenbahntechnische Rundschau, August, S. 381 und 383.
Schleifzug zur Beseitigung von Schienenriffeln. (800 Wörter & Abb.)
Schienenschleifwagen der Rhätischen Bahn zur Riffelbeseitigung. (1 000 Wörter & Abb.)

Elektrische Bahnen. (München.)

1955 **625** .258
Elektrische Bahnen, Juli, S. 145.
LENK (H.). — Die **Ansprechzeit von elektromagnetischen Schienenbremsen.** (8 000 Wörter & Abb.)

1955 **621** .431.72 (436)
Elektrische Bahnen, Juli, S. 155.
ZILKA (O.). — Die **neuen dieselelektrischen Bo'Bo'-Lokomotiven** der Österreichischen Bundesbahnen, Reihe VL 2045, 1000 PS. (5 000 Wörter & Abb.)

Glasers Annalen. (Berlin.)

1955 **621** .138.5
Glasers Annalen, Mai, S. 129.
GRÖNEGRESS (H.W.). — **Einsparungen durch Brennhärten bei der Instandhaltung von Dampflokomotiven.** (5 000 Wörter & Abb.)

1955 **621** .7 : **621** .335
Glasers Annalen, Mai, S. 139 und 143.
WINKELMANN (R.). — Erfahrungen mit dem **Brennhärten in der Einzelfertigung von Bauteilen für Elektro-Lokomotiven.** (2 000 Wörter & Abb.)
BUSSMANN (W.). — **Erfahrungen mit brenngehärteten Ritzeln für Achsantriebe** von elektrischen Lokomotiven. (1 500 Wörter & Abb.)

1955 **621** .135.2 & **625** .212
Glasers Annalen, Mai, S. 146.
OVERKOTT (F.). — Stand der **Spurkranzhärtung an Lokomotiv- und Waggonradsätzen.** (2 000 Wörter, 2 Tafeln & Abb.)

1955 **625** .15
Glasers Annalen, Mai, S. 150 und 155.
STEIGER (H.). — **Brennvergüten von Schienen für Weichen und Kreuzungen und von Schienenenden.** (2 000 Wörter & Abb.)
ISENBERG (H.). — Versuchsergebnisse und Erfahrungen beim **Brennvergüten von Weichenteilen.** (1 000 Wörter & Abb.)

1955 **621** .33 (469)
Glasers Annalen, Mai, S. 160.
Portugal elektrisiert seine Eisenbahn mit 50 Hz. (300 Wörter & Abb.)

Internationales Archiv für Verkehrswesen. (Mainz.)

1955 **656**
Int. Archiv. für Verkehrswesen, Nr. 12, 2. Juniheft, S. 265.
FEINDLER (R.). — Die **Wahl des Verkehrsmittels** vom Verkehrsnutzer gesehen. (8 000 Wörter.)

1955 **656**
Int. Archiv. für Verkehrswesen, Nr. 12, 2. Juniheft, S. 271; Nr. 13, 1. Juliheft, S. 297.
LINDEN (W.). — Welche Faktoren beeinflussen die **Wahl des Verkehrsmittels?** (10 000 Wörter.)
PENTINGA (K.J.). — Welche Faktoren beeinflussen die **Wahl des Verkehrsmittels?** Neue Wege zum Schutz der Transportgüter. (2 000 Wörter.)

1955 **656** .225 & **656** .261
Int. Archiv. für Verkehrswesen, Nr. 12, 2. Juniheft, S. 280.
ELLERSIEK (K.). — Konstruktive und bauliche Möglichkeiten zur **Ausdehnung des Haus-Haus-Verkehrs.** (4 000 Wörter.)

1955 **656** .225
Int. Archiv. für Verkehrswesen, Nr. 13, 1. Juliheft, S. 289.
HEGNER (F.). — **Koordination und Mechanisierung im Güterumschlag** und Senkung der Umschlags- und Transportkosten. (4 000 Wörter & Abb.)

Signal und Draht. (Frankfurt a. Main.)

1955 **656** .251
Signal und Draht, September, S. 137.
GRANDRATH (F.). — Anwendung und Anordnung des **Grundsignals.** (2 500 Wörter & Abb.)

1955 **656** .254
Signal und Draht, September, S. 141.
AST (W.). — Über das **Registrieren der Belegung von Bündeln in Fernmeldenetzen.** (2 000 Wörter & Abb.)

1955 **656** .259
Signal und Draht, September, S. 146.
LATSCH (A.). — Hilfsgerät beim Einbau von **Schienenkontakten.** (600 Wörter & Abb.)

Verkehr. (Wien.)

1955 **656** .235 (494)
Verkehr, 30. Juli, S. 969.
DIRLEWANGER (H.). — **Probleme der künftigen schweizerischen Gütertarifpolitik.** (1 000 Wörter.)

1955 **656** (436)
Verkehr, 6. August, S. 997.
Koordinierung Schiene-Strasse oder Ordnung des Strassenverkehrs? Eine Stellungnahme der Generaldirektion der ÖBB. (1 000 Wörter.)

In English.

The Engineer. (London.)

1955 **621** .132.1 (42)
The Engineer, August 12, p. 230.
POULTNEY (E.C.). — **Notable locomotives of 1905.** (4 800 words & figs.)

Engineering. (London.)

1955 **621** .431.72 (41)
Engineering, August 5, p. 185.
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1955 **621** .33
Engineering, August 12, p. 209.
ANDREWS (H.I.). — **Heat losses of locomotive boilers.** Measuring the effects of radiation and exhaust-steam injectors. (1 800 words & figs.)

1955 **621** .438 (42)
Engineering, August 19, p. 239.
FITTON (A.) and VOYSEY (R.G.). — **Solid-fuel-fired gas turbine in Great Britain. The design of pulverised-coal burning systems.** (6 000 words & figs.)

Proceedings of the Institution of Mechanical Engineers. (London.)

1954 **669** .1
Proceedings of the Institution of Mechanical Engineers, Volume 168, Number 16, p. 465.
BARDGETT (W.E.). — **Comparative high-temperature properties of British and American steels.** (4 000 words, tables & figs.)

1955 **621** .438
Proceedings of the Institution of Mechanical Engineers, Volume 169, Number 7, p. 163.
MORDELL (D.L.). — **An experimental coalburning gas turbine.** 18 pages illustrated.)

The Locomotive. (London.)

1955 **625** .216
The Locomotive, July, p. 111.
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1955 **621** .431.72 (494)
The Locomotive, July, p. 121.
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1955 **621** .431.72 (41)
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1955 **621** .338
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1955 **621** .431.72 (6)
The Locomotive, August, p. 131.
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Modern Railroads. (Chicago.)

1955 **625** .171 (73)
Modern Railroads, July, p. 51.
Ultrasonic rail test car. (300 words & figs.)

1955 **621** .33 (73)
Modern Railroads, July, p. 52.
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Modern Transport. (London.)

1955 **621** .431.72 (494) & **625** .3 (494)
Modern Transport, July, 2, p. 6.
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New railcars for narrow gauge railways. (1 200 words & figs.)

1955 **621** .431.72 (42)
Modern Transport, July 23, p. 3; July 30, p. 12.
Diesel engine range for rail traction. The Paxman welded frame V series YL type. (2 300 words & figs.)

1955 **621** .431.72 (41)
Modern Transport, July 30, p. 5.
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1955 **621** .132.8 (6)
Modern Transport, August 27, p. 12; September 3, p. 5.
New locomotives for East Africa. (1 700 words & figs.)

1955 **621** .431.72 (42)
Modern Transport, August 27, p. 15.
Diesel-electric shunters for British Railways. (1 200 words & figs.)

The Oil Engine and Gas Turbine. (London.)

1955 **621** .431.72 (41)
The Oil Engine and Gas Turbine, August, p. 122.
C.I.E's large programme for locomotives. (1 200 words & figs.)

Railway Age. (New York.)

1955 **625** .28
Railway Age, June 20, p. 64.
BLEIBTREU (H.). — **Lightweight trains.** — Where they should be used. — How they should be built. (2 600 words & figs.)

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1955 **625** .172 (73)
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1955 **625** .13 (44)
The Railway Gazette, July 29, p. 129.
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Diesel Railway Traction
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1955 **621** .431 .72 (485)
Diesel Railway Traction, July, p. 223.
Swedish bogie gear-drive railbuses. (1 900 words & figs.)

1955 **621** .431 .72 (44)
Diesel Railway Traction, August, p. 229.
Tests of French high-speed engine. (2 700 words, tables & figs.)

1955 **621** .431 .72
Diesel Railway Traction, August, p. 243.
Main-line Diesel traction. An examination of certain aspects which become vital on the decision to go ahead with any large-scale programme such as that now proposed for British Railways. (2 800 words.)

1955 **621** .431 .72 (492)
Diesel Railway Traction, August, p. 247.
Freight locomotives for Holland. (1 400 words & figs.)

Railway Locomotives and Cars. (New York.)

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Railway Locomotives and Cars, July, p. 47.
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1955 **621** .431 .72 (73)
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The Railway Track and Structures. (Chicago.)

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The Railway Track and Structures, August, p. 22.
How to raise track-fast! ...« Sleds » make it a downhill job. (3 200 words & figs.)

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Ferrocarriles y Tranvias. (Madrid.)

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Ferrocarriles y Tranvias, abril, p. 94.
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1955 **625** .25
Ferrocarriles y Tranvias, abril, p. 98.
JOHANSSON (A. V.) & STIGER (H. R.). — El **frenado eléctrico en las locomotoras Diesel eléctricas.** (3 000 palabras & fig.)

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Ferrocarriles y Tranvias, mayo, p. 116.
RODRIGUEZ del PALACIO (G.). — Consideraciones sobre la **protección de pasos a livel.** (2 500 palabras & fig.)

1955 **625** .144 .4
Ferrocarriles y Tranvias, mayo, p. 127.
Limpiadora de balasto para los Ferrocarriles británicos. (600 palabras & fig.)

Revista de Obras Públicas. (Madrid.)

1955 **621** .33 (460)
Revista de Obras Públicas, agosto, p. 391.
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1955 **62** (01
Revista de Obras Públicas, agosto, p. 416.
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1955 **625** .172
Ingegneria Ferroviaria, luglio-agosto, p. 553.
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1955 **625** .143 .3
Ingegneria Ferroviaria, luglio-agosto, p. 562.
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Rivista di Ingegneria. (Milano.)

1955 **62** (01
Rivista di Ingegneria, agosto, p. 927.
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1955 **656** .
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CASCINO (C.). — Elementi per lo studio dell **ammodernamento dei mezzi di trasporto su rotaie.** La statistica del traffico e i costi di esercizio. (10 000 parole tavole & fig.)

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De Ingenieur. (Den Haag.)

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BIJL (F.), DOUZE (H.) & DE PATER (A. D.). — **Beproevingsinrichting voor Dieselmotoren.** Funderingen der proefstanden en bepaling van de door de motoren afgegeven arbeid. (1 000 woorden.)

1955 **621** .43
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Inženýrské Stavby, nº 5, May 19, p. 195.

KLIMES (J.) & SMITKA (V.). — Remarkable **construction of a concrete bridge** by the Czechoslovakian Railways. (3 000 words & figs.)

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MILIK TICHY. — **Statically indeterminate construction in prestressed concrete.** (3 000 words & figs.)

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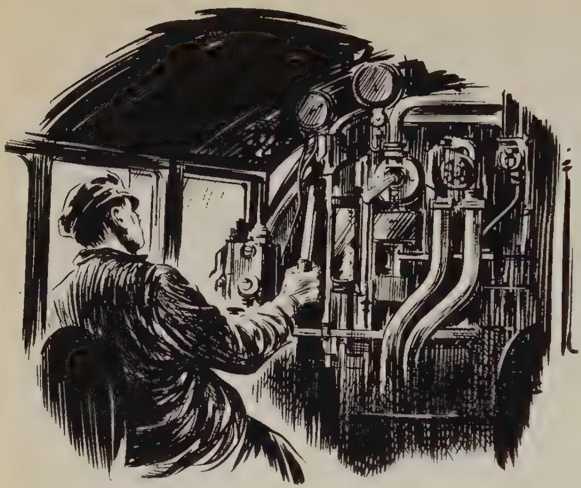

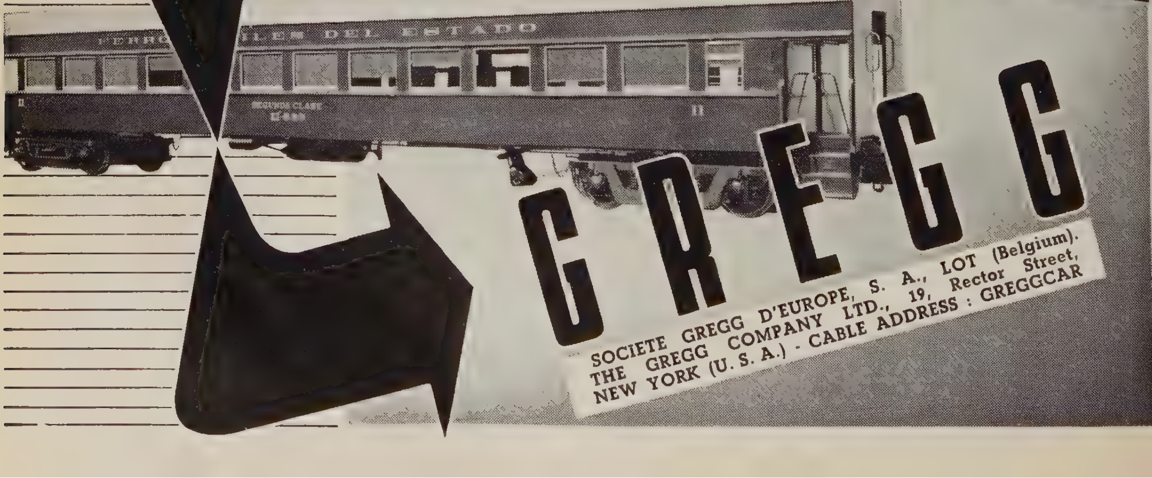

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